Intermediate Vibration

Course 2032

Prepared by
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Sincerely,

Greg Carr
Director, Reliability Solutions Global Services and Knoxville Site Leader

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Appendix A • Glossary

Basic Vibration and Data Analysis Review

Section 1

Objectives

- Define vibration.
- Describe the different methods of measuring vibration.
- Understand Amplitude and Frequency.
- Examine amplitude measurements.
- Describe the Time Waveform Data Display.
- Identify different Waveform Characteristics.
- Describe the Spectral Data Display.
- Identify Spectral Data Characteristics.
- Describe the steps of Spectral Fault Analysis.

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General Description

A variety of different parameters can be measured for rotating equipment. Examples of measurable parameters include: pressure, temperature, flow, density, voltage, speed, torque and viscosity. Of all the parameters that can be measured, the vibration signature contains the most information. The vibration signature not only provides information concerning the severity of a problem, but it also points to the possible source of a problem.

The study of the vibration signature begins with the use of a non-intrusive tool to measure mechanical vibration. This signature provides information concerning the severity of a problem. With the right tools and careful analysis, it is also possible to determine potential **sources** of the problem.

What is Vibration?

Vibration is the movement of a body about a reference. The reference is the position of the mass when it is at rest, with no outside forces acting upon it. In rotating machinery, the center-line of the shaft is the reference. Objects that are vibrating move back and forth, from one position to another.

Vibration occurs when an object responds to some form of excitation or force. The **excitation** is generally referred to as a **forcing function**. If the forcing function is constant or continuous, then the object or system will continue to vibrate until the forcing function stops.

Forcing Functions can be periodic or random. **Periodic vibration** is movement occurring in a repetitive manner. **Random vibration** is considered to be non-periodic. Random forces occur as transient or momentary events.

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For most industrial machinery, some potential **forcing functions** include:

- Imbalance of rotating components
- Misalignment of couplings, bearings, gears and sheaves
- Defects and deterioration of Rolling Element Bearings
- Gear wear
- Belt Defects
- Electrical problems (imbalanced magnetic forces) in motors
- Hydraulic Instabilities in Journal Bearings
- Aerodynamic/Hydraulic forces in fans, blowers, and pumps
- Eccentricity of rotating components such as V-belt pulleys or Gears

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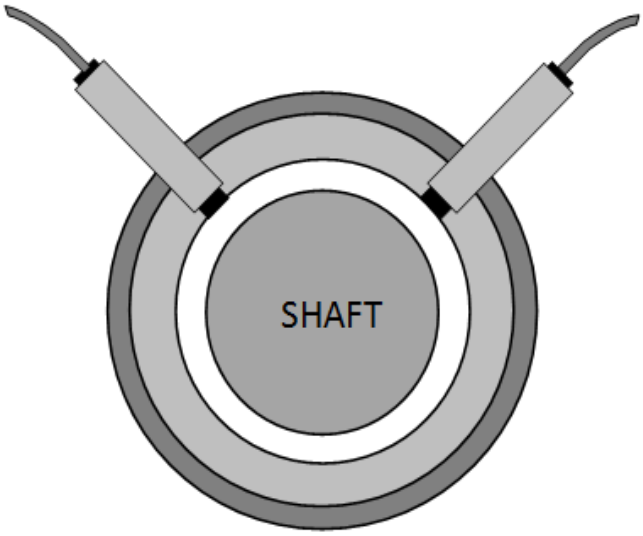
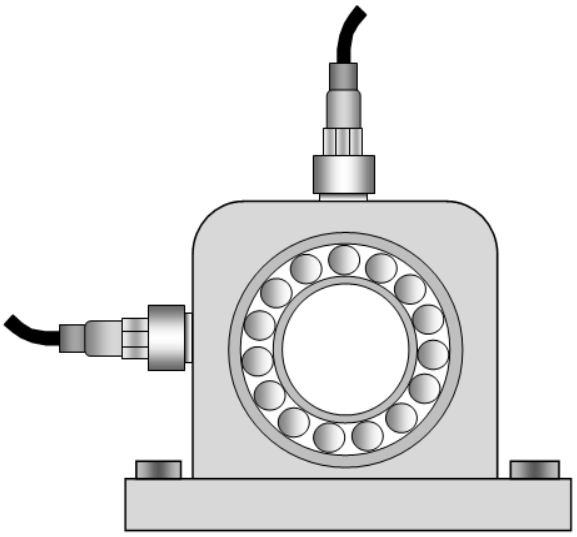
How is Vibration Measured?

Vibration is measured using different types of vibration transducers. Transducer types and functions are discussed in Chapter 3 - Data Collection.

Vibration is measured on rotating equipment using two methods or techniques. One method measures the actual movement of the shaft within the casing as shown below. This is the preferred method of data collection on sleeve bearings.



Another method, as shown below, is to measure the vibration of the casing itself. This method is preferred on anti-friction bearings.



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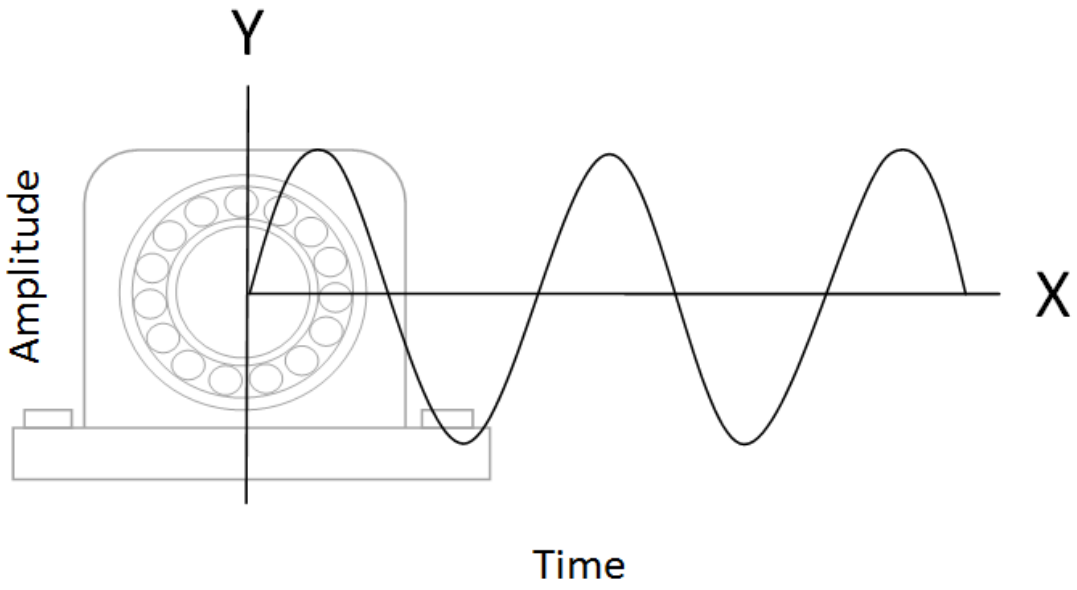
Time Domain – Waveform

The transducers convert mechanical energy into an electrical signal (a number value). The number values are plotted over time. The plotted movement is illustrated in a **time waveform**. The time waveform is a graphical representation of how much a system moves over a period of time. It is defined as a display of **amplitude vs. time**.

The figure below illustrates the measured movement of a bearing housing. The overlying plot is a result of that movement. This plot resembles a sine wave.



The **Y axis** of the waveform displays the **amplitude**, or the measured amount of the movement.

The **X axis** of the waveform represents the amount of **time** it took to move. This value is usually presented in seconds and sometimes in milliseconds.

The amount of information that a waveform contains will depend in part upon the length or total amount of time in the waveform. The total amount of time in the waveform is typically referred to as the **time record**. A typical time record for a normal vibration measurement is usually no more than a few seconds. Sometimes more time is required for the measurement depending on the forcing function and type of faults suspected.

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Amplitude

The strength of the vibration signal is displayed as the amplitude.

The amplitude (maximum extent or magnitude) of vibration is measured from the time waveform. Measuring and analyzing the **amplitude** of the vibration will help determine the **severity** of a particular vibration.

Higher amplitudes indicate more movement or more stress experienced by the machine; the higher the vibration amplitudes, the greater the **severity** of the vibration. It is the measured vibration amplitude that usually determines if a machine's vibration is normal and acceptable or if the vibration levels warrant a closer inspection or possibly a shut down for repair.

Amplitude Measurements

When measuring vibration, the original measurement will represent one of the following three amplitudes:

- **Displacement -** The total amount of back and forth movement.
- **Velocity -** The average speed of the back and forth movement.

Acceleration - The rate of change in velocity - the force of the movement.

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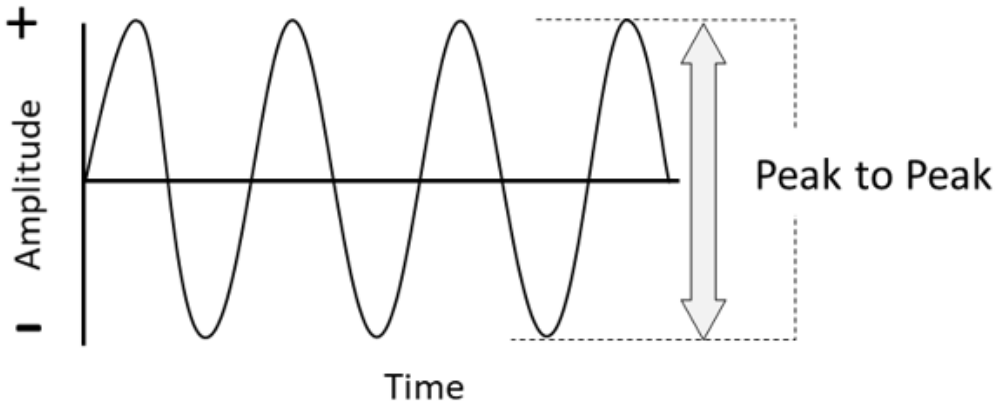
Displacement

The total distance a body, or mass, travels in relation to a reference point. (Peak to Peak or Pk–Pk)

- The total movement is measured from the positive (+) peak value to the negative (-) peak value. Therefore, displacement is a Peak to Peak value.
- Displacement measurements are commonly expressed in units of mils (1 mil = .001 inch) or microns (1 micron = .001 millimeter).



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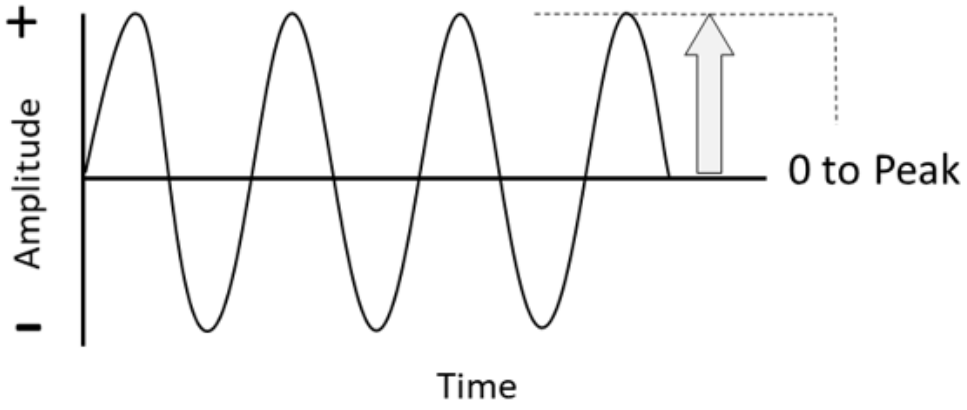
Velocity

The rate at which displacement occurs. (Peak or Pk)

- Measured from the at-rest position (0) to the peak value (maximum positive position).
- Displacement of the machine, or component, in relation to time. As a vibration measurement, this is a Peak value.
- Velocity measurements are commonly expressed in units of inches per second (in/sec) or millimeters per second (mm/sec).



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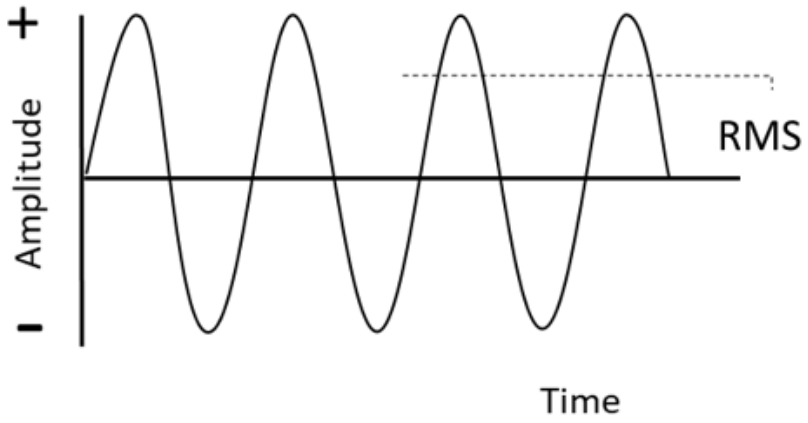
Acceleration

The change in velocity per unit time; total force acting on a body.

- Acceleration is the change in velocity with respect to time. Acceleration is measured from the time waveform as the amount of energy in the area under the curve. This is an RMS (root mean square) measurement.
- When applied to a dynamic signal such as vibration, RMS refers to the magnitude of a signal obtained by averaging the squared values of the vibration signal over a period of time, then taking the square root of the result.
- Acceleration is expressed in units of G's (1g = 386 in/sec)^2.



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Frequency

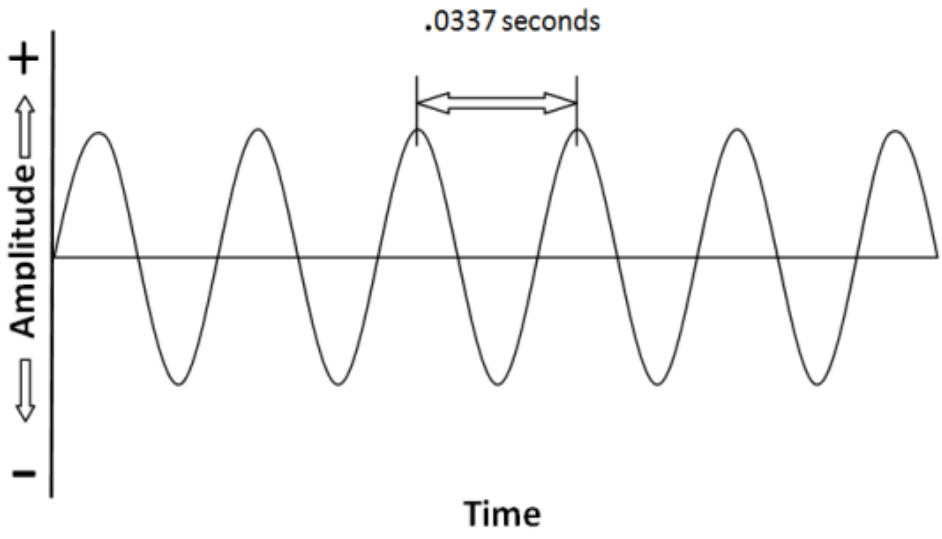
Frequency is defined as how often an event occurs per unit of time.

As a machine vibrates, different forcing functions cause different rates of movement. These repetitive (periodic) cycles of movement can be slow, or fast, depending on the forcing function.

The rate at which a bearing housing is vibrating is called a vibrating frequency. This movement is typically repetitive. Repetitive cycles of movement occurring at a fast rate are considered to be high frequency vibration. A cycle of movement at a slow rate would be considered low frequency vibration.

Frequency is calculated from the period of a cycle in the waveform. A period in the waveform is the amount of time it takes to complete one cycle of vibration. **Frequency (F)** is the inverse of the **time period (T)**.

For example, if one cycle of vibration takes .0337 seconds to complete, then the time period of that cycle is .0337 seconds. To determine the frequency of that period, divide 1 by the time period. In this example, 1 is divided by .0337 to find the frequency of the cycle.



$$F = 1/ T$$

$$F = 1/ .0337$$

$$F = 29.7$$

Therefore, this cycle repeats itself 29.7 times every second. (29.7 Hz.)

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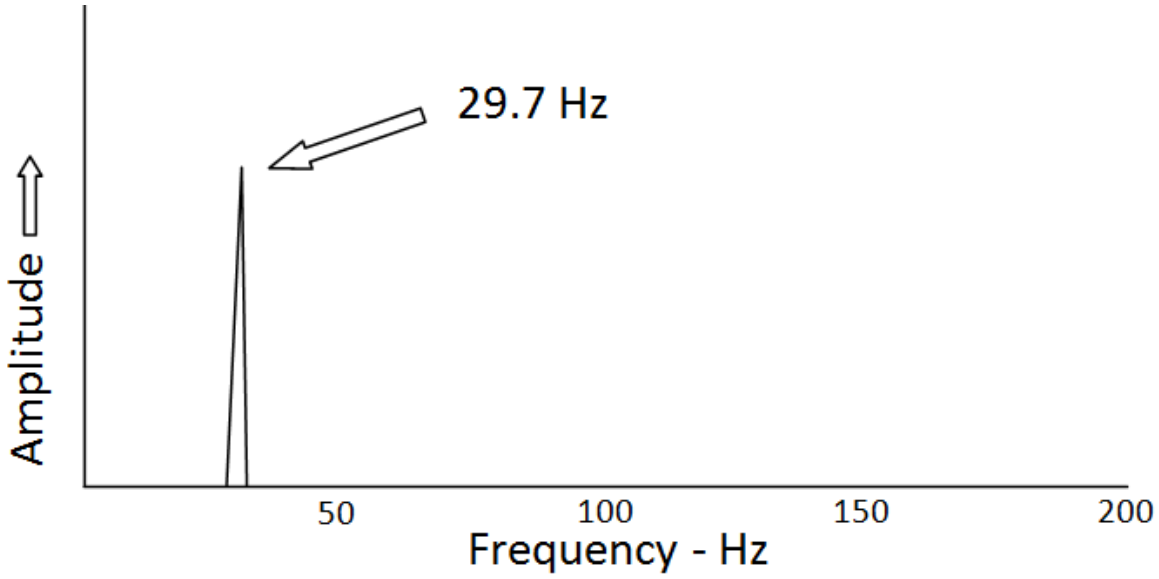
Frequency Domain

Once the frequency of a vibration cycle is calculated, it can be plotted in the frequency spectrum. This transition is accomplished by the Fast Fourier Transform, which is discussed in detail later in this section.

The **spectrum** represents the amplitude and the frequency of the periodic cycles from the waveform.

The three amplitude units used in the waveform also apply to the frequency spectrum.

The vibration cycle of .0337 from the previous waveform example is now plotted as a frequency of 29.7 Hz in the spectrum as seen below.



The peak in this spectrum represents the amplitude and frequency of the .0337 second cycle from the waveform.

Like the waveform, the frequency spectrum is displayed as an **X - Y plot**.

The **Y axis** of the spectrum displays the **amplitude**, or the measured amount of the movement.

The **X axis** of the spectrum represents the **frequency**, or how often the cycle of vibration is occurring.

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Frequency Units

Frequency may be expressed three different ways:

- **Cycles per minute** (CPM) – how many times a cycle occurs in one minute.
- **Cycles per second** (CPS or Hz) – how many times a cycle occurs in one second.
- **Orders** – how many times a cycle occurs in one shaft revolution.

Orders can be described as the relationship of any frequency to shaft turning speed.

Hertz (Hz) is an expression of cycles per second and is typically used instead of CPS.

To convert any frequency from CPM to Hz, divide CPM by 60, since there are 60 seconds in one minute. To convert from Hz to CPM, **multiply** the value by 60.

For example: If a shaft is rotating at a frequency of 1785 cycles per minute (CPM), it is also accurate to say that the shaft is rotating at a frequency of 29.75 cycles per second (CPS or Hz).

Examples of CPM and Hz:

•1785 CPM/ 60 = 29.75 Hz

•29.75 x 60 = 1785 CPM

•3550 CPM/ 60 = 59.17Hz

•59.17 x 60 = 3550 CPM

To determine the **order** of a frequency, divide any frequency by the rotating frequency (turning speed) of the shaft being measured, using the same frequency units.

For example: The shaft turning speed of a machine is 1785 RPM (rotations per minute). A vibration frequency appears in the spectrum at 1785 CPM. To find the order of 1785 CPM, divide 1785 CPM by 1785 RPM. In this case, the Order is 1.

•1785 CPM/ 1785 RPM = 1 Order

Using the same turning speed in Hz and RPS (rotations per second), the results are the same.

•29.75 Hz/ 29.75 RPS = 1 Order

•(Many times expressed at 1 x Turning Speed or 1 x TS.)

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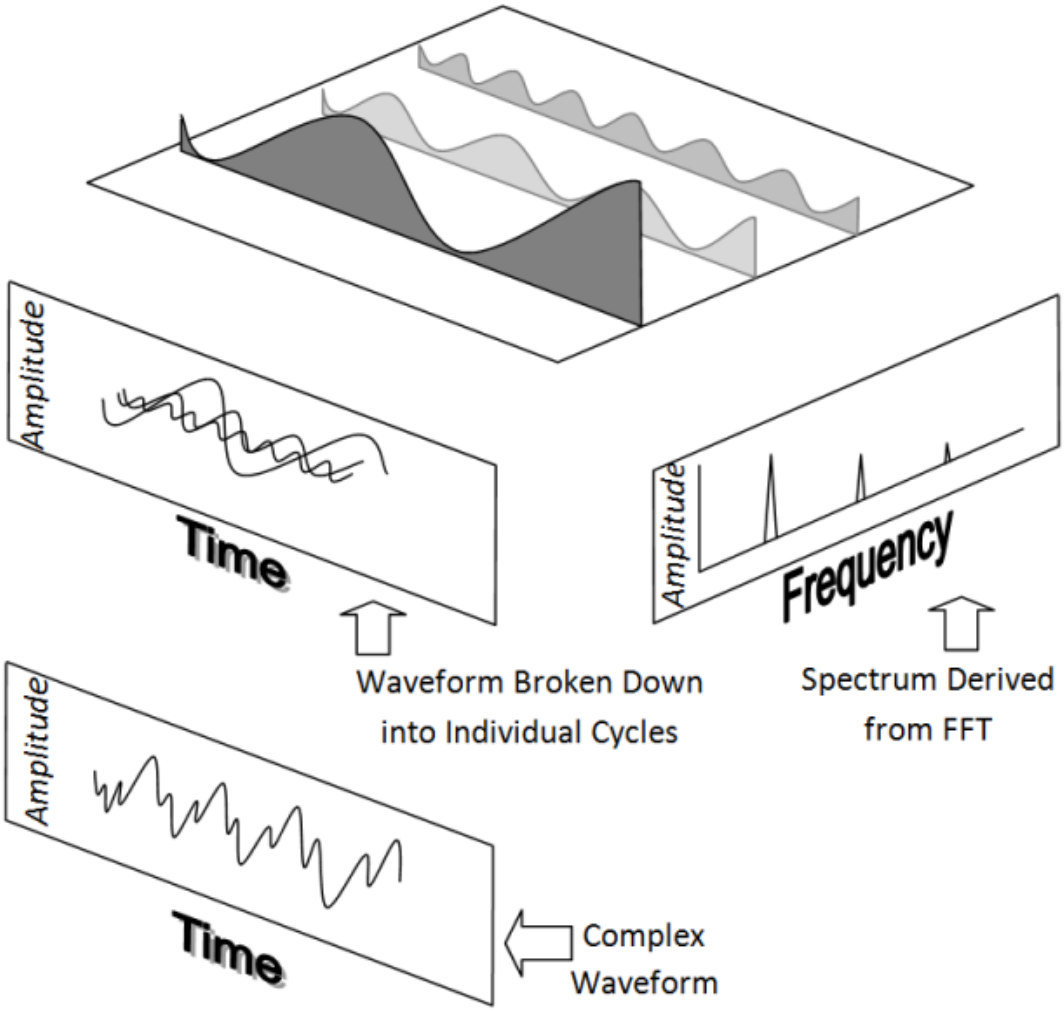
FFT – Fast Fourier Transform

Machine vibrations usually consist of many vibratory movements from different forcing functions within the equipment train such as bearings, gears, belts, pumps, fans, as well as vibrations from nearby equipment. These various vibrations will be present in a single measurement, resulting in a very complex waveform.

In the complex waveform, the individual frequencies of vibration from multiple components in the machine are not clearly displayed. Due to the complexity of the waveform, diagnosing a machine problem and determining the source of forcing function from the waveform is difficult.

The transition from the time domain to frequency domain spectrum is accomplished by the **Fast Fourier Transform (FFT)**. A graphic depiction of the mathematical process is shown below.



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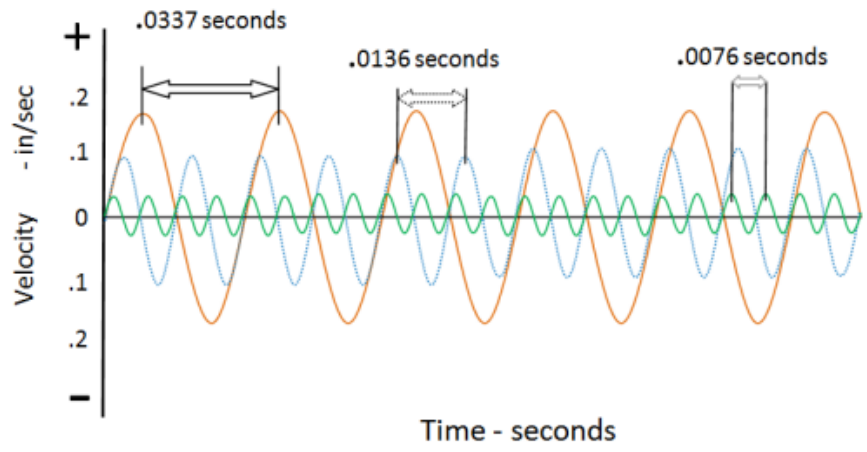
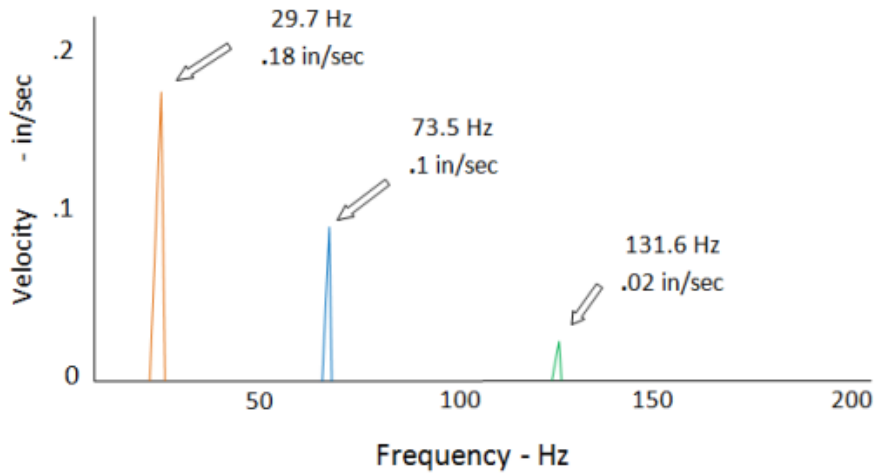
As a result of the FFT process, the frequency spectrum will contain peaks representing the amplitude and frequency of each periodic cycle of vibration in the waveform. The relationship of the peaks in the spectrum can then be related to individual machine component vibrations.

Therefore, the spectrum (not the waveform) becomes the primary choice for analyzing machine vibration, with the exception of certain types of faults.

In the example waveform below, there are three distinct periodic cycles of vibration. In this example, the amplitude represents velocity for both the waveform and the spectrum.



Using the expression F =1/ T, the frequencies are calculated from the waveform and displayed in the spectrum below. In this example, the frequency units are in Hz (Cycles per Second).



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Amplitude Relationships and Conversion

Amplitude is the measurement of the energy or movement of a vibrating object. The change in amplitude corresponds with the change in the severity of the problem. Conversion factors for the three units of amplitude are shown below.

RMS	Root Mean Square	0.707 times the true peak value
PK-PK	Peak-to-Peak	2 times the true peak value
PK	Peak	1.414 times the rms value

Amplitudes may be manually converted from one unit to the other using the correct equations under certain conditions. These equations are **frequency specific** and must be applied to **sinusoidal** waves only. They are not intended for converting overall amplitudes.

- $V = 0.0031416 \times f \times D$
- $V = (86.75 \times A) \div f$
- $A = 0.01146 \times V \times f$
- $A = 0.00003613 \times D \times f^2$
- $D = (318.47 \times V) \div f$
- $D = (27668 \times A) \div f^2$

A = Acceleration in g's – RMS

V = Velocity in in/sec – Peak

D = Displacement in mils – Peak to Peak

f = Frequency in Hz

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Example Calculation:

Assume the vibration amplitude is 6 Mils at 1785 CPM and convert to Velocity.

Start with:

$$V = 0.0031416 \times f \times D$$

$$V = 0.0031416 \times 29.75 \times 6$$

$$V = 0.56 \text{ in/sec}$$

Note

These equations are **FREQUENCY SPECIFIC**. They must **NOT** be used to convert overall amplitudes.

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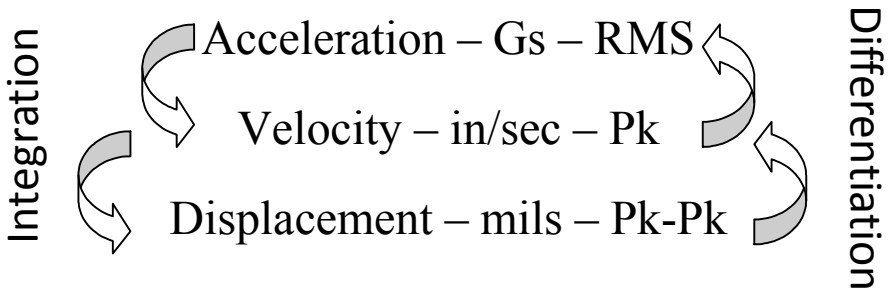
Integration and Differentiation

The different amplitude measurements are mathematically related to one another using processes referred to as **Integration** and **Differentiation**.

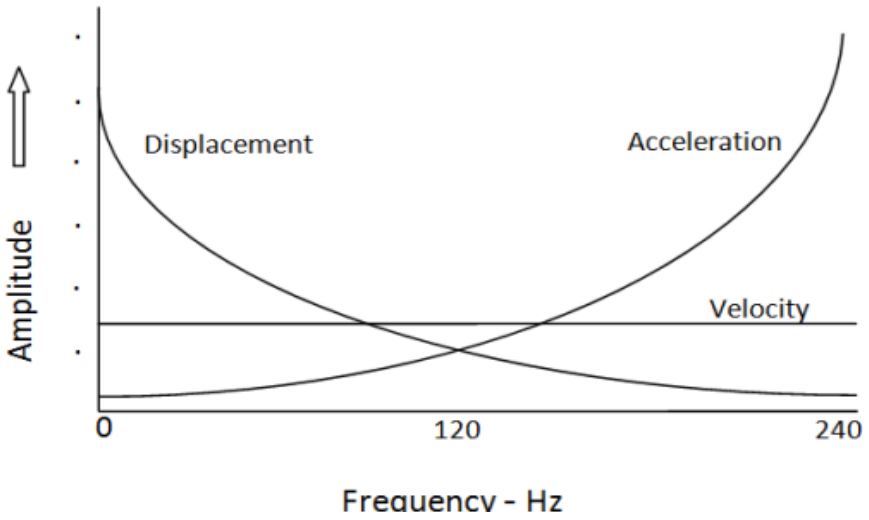
Integration is the mathematical conversion of Acceleration to Velocity, and/or Velocity to Displacement.

Differentiation is the mathematical conversion of Displacement to Velocity, and/or Velocity to Acceleration.

Integration is preferred over Differentiation when converting vibration amplitudes.



As seen in the plot below, when comparing the three amplitude units in the frequency domain, velocity becomes the amplitude unit of choice for spectral analysis since velocity is not influenced by frequency. Displacement has the effect of enhancing lower frequencies, but suppressing higher frequencies. Acceleration has the effect of enhancing higher frequencies, while suppressing lower frequencies.



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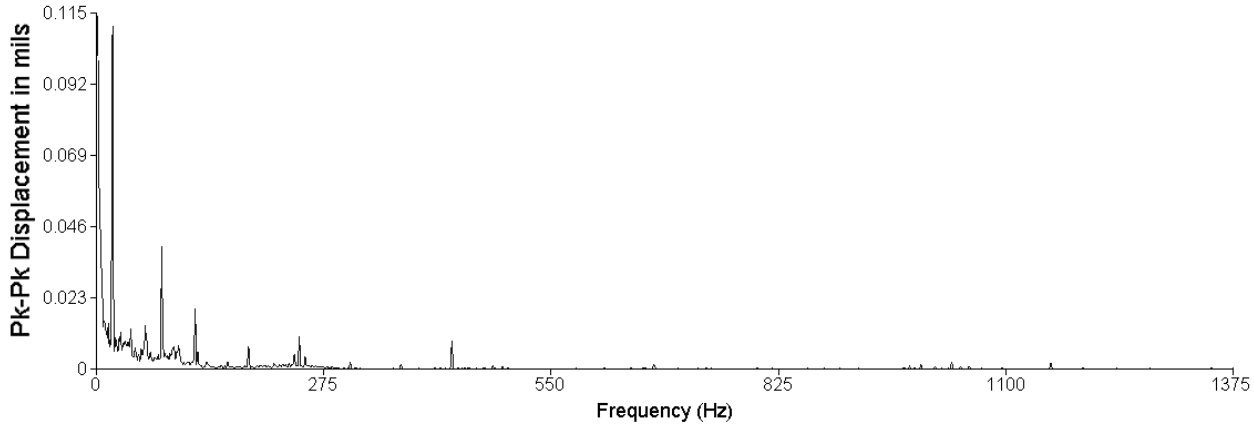
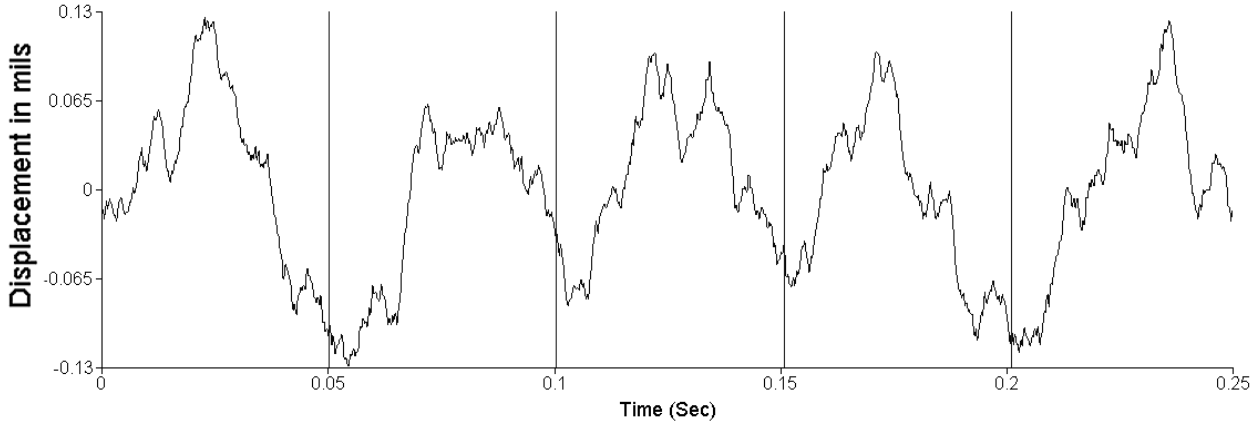
Amplitude Relationships - Displacement Data





The spectrum and waveform above were acquired in units of G's - Acceleration. The data has been double integrated to displacement.

When viewed in displacement, amplitude values at higher frequencies are attenuated (reduced) in both the spectrum and waveform.

There is less displacement (overall movement) to measure at higher frequencies; therefore the displacement values have much lower amplitudes in the high frequency range when viewing the spectrum in displacement.

When viewed in displacement, the waveform may have, to some extent, a sinusoidal pattern, with very few high frequency cycles of energy visible.

Displacement is the preferred amplitude when looking for faults and machine characteristics that show up in the lower frequency range of the spectrum.

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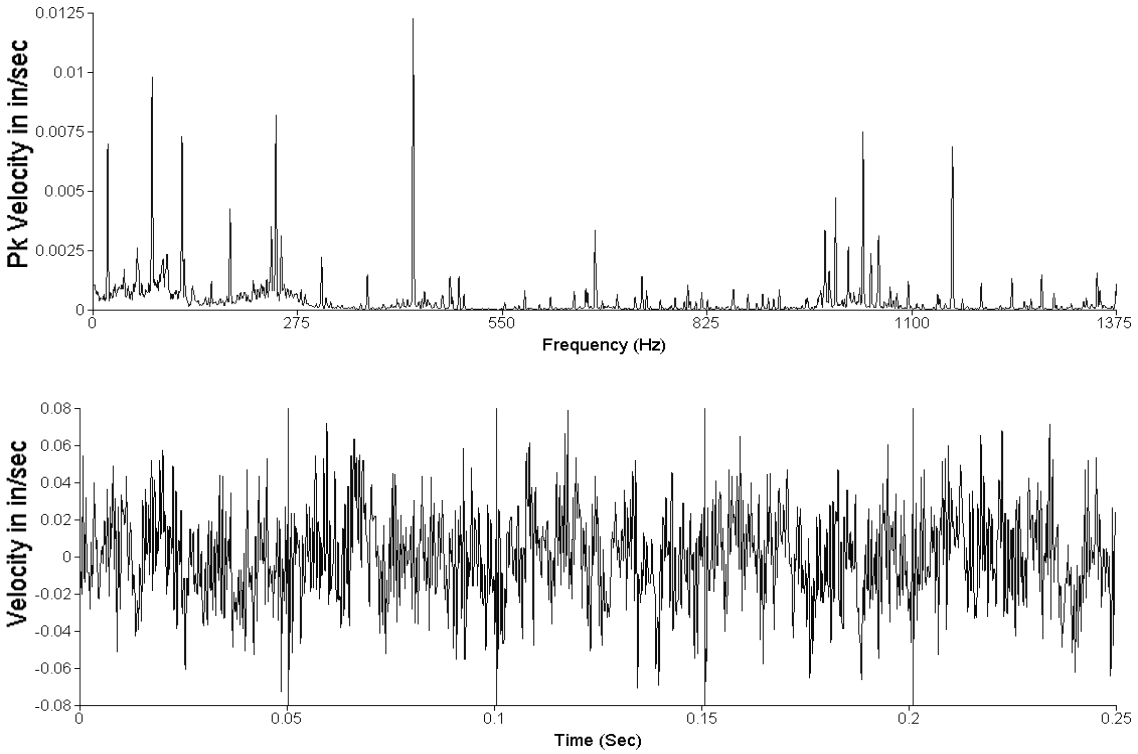
Amplitude Relationships - Velocity Units



The same vibration data from the previous plots are now displayed in units of in/sec - Velocity.

This velocity spectrum clearly shows both low and high frequency peaks, many that were not present in the displacement spectrum.

Viewing the above velocity spectrum, there is a peak around 430 Hz that appears to be near .0125 in/sec (higher than most other peaks in the spectrum). Since velocity treats amplitude equally across the frequency spectrum, the highest vibration in this measurement appears at 430 Hz with an amplitude of .0125 in/sec.

The velocity waveform shows higher frequency energy that was not present in the displacement waveform. However, an indication of the sinusoidal pattern is still present.

Velocity allows the analyst to compare amplitude values at all frequencies present in the spectrum and is therefore the preferred amplitude unit when viewing spectral data.

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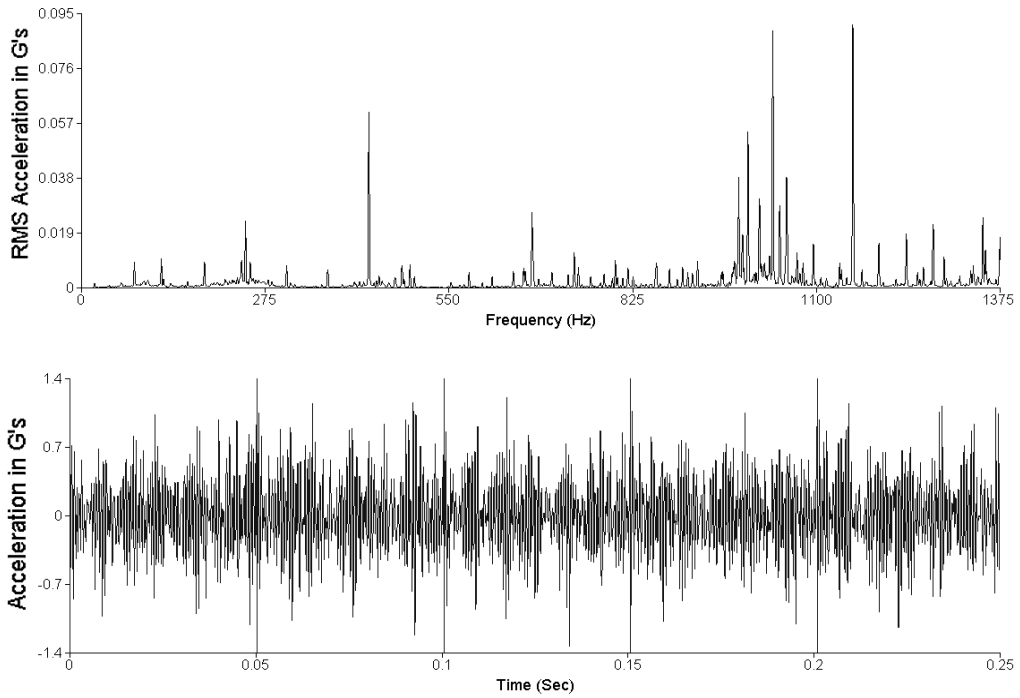
Amplitude Relationships - Acceleration Data



Here are plots of the same vibration measurement as before, now in units of Acceleration.

Acceleration is a measurement of the force behind the forcing function that is causing the vibration.

There is less acceleration (force) to measure at lower frequencies than at higher frequencies; therefore the acceleration amplitude values will be low in the lower frequency range of the acceleration spectrum.

In the acceleration waveform, the higher frequency energy is much more evident. Since the amplitude of force is a very good indication of the severity of the problem, acceleration is the amplitude of choice when viewing the time waveform.

Any of the three amplitude units can be used during data analysis of both the spectrum and waveform. Different fault characteristics within the machine will require the analyst to view the data in different amplitude units.

Note

The analyst should always be aware of the amplitude units being displayed when viewing any vibration plot.

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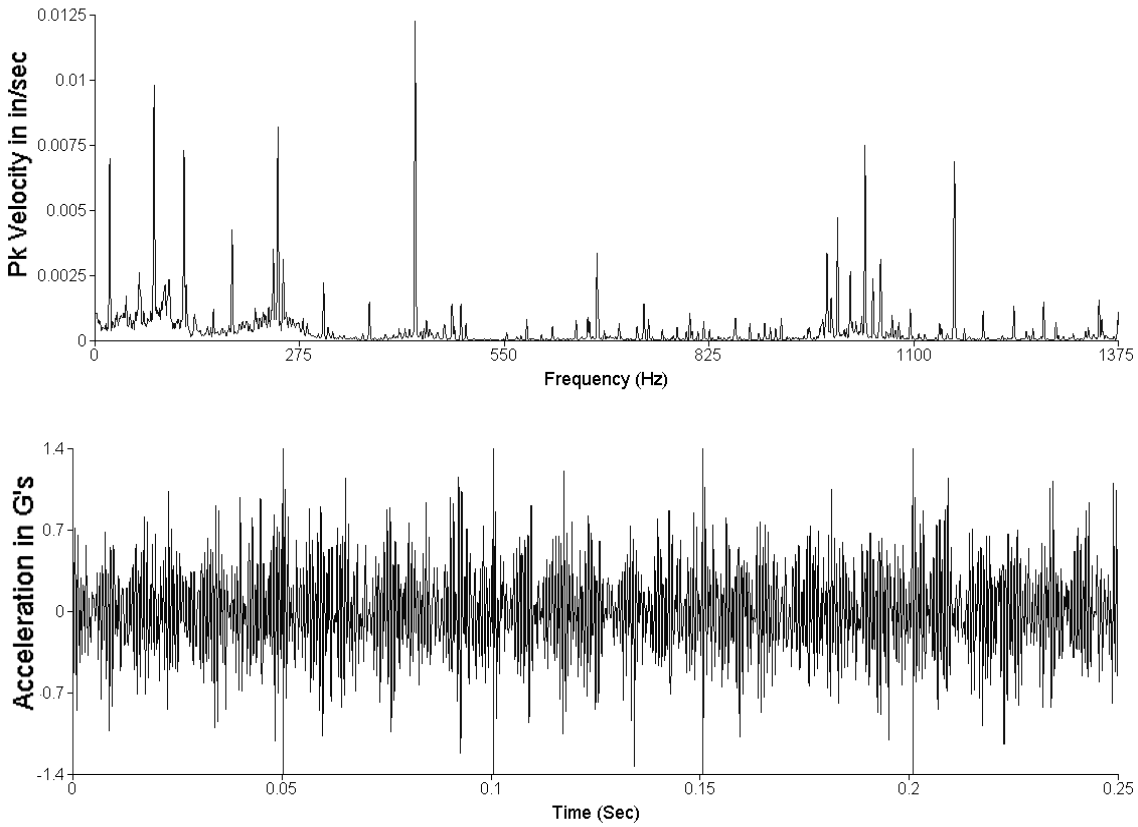
Preferred Amplitude Units for Machine Analysis

The following spectrum and waveform are displayed in the preferred units for viewing and analyzing most machine vibration data.



The spectrum is in units of Velocity and the waveform is in units of Acceleration.

The **spectrum** is typically stored and displayed in units of **velocity**. Velocity treats amplitude equally across the frequency domain, which allows an accurate comparison of the peaks present in the spectrum. During analysis, the amplitude units may be changed at any time to displacement and/or acceleration to allow lower and higher frequencies to be viewed more closely.

The **waveform** is usually stored and displayed in units of **acceleration**. Like the spectrum, the waveform amplitude units may be changed at any time during analysis, depending on the type of faults that are suspected.

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Review of Amplitude and Frequency Units

AMPLITUDE

- Acceleration (G's – RMS)
- Velocity (in/sec – peak)
- Displacement (Mils – peak to peak)

Changing amplitude units has a significant effect on the appearance of the data with respect to low frequency vs. high frequency peaks and amplitudes.

FREQUENCY

- Cycles per Minute (CPM)
- Cycles per Second (Hz or CPS)
- Orders (Given Freq* / TS in RPM*)

To avoid confusion, the units for each variable should be the same. (Hz and Hz or CPM and CPM)

Frequency unit selection can be important. Occasionally, viewing data in Orders vs. Hz or CPM makes analysis easier, depending on the defect. Analysts should be familiar with the way peaks are labeled and how the cursor information is displayed in the data using the various units. Examples using different frequency units are presented later in this course.

Note

The terms RPM and CPM are often used interchangeably. In most cases however, a CPM frequency will not equal an RPM frequency. CPM is **Cycles** of vibration per minute. RPM is **Rotations** of the shaft per minute. There are other examples of this difference that will be covered later in the course.

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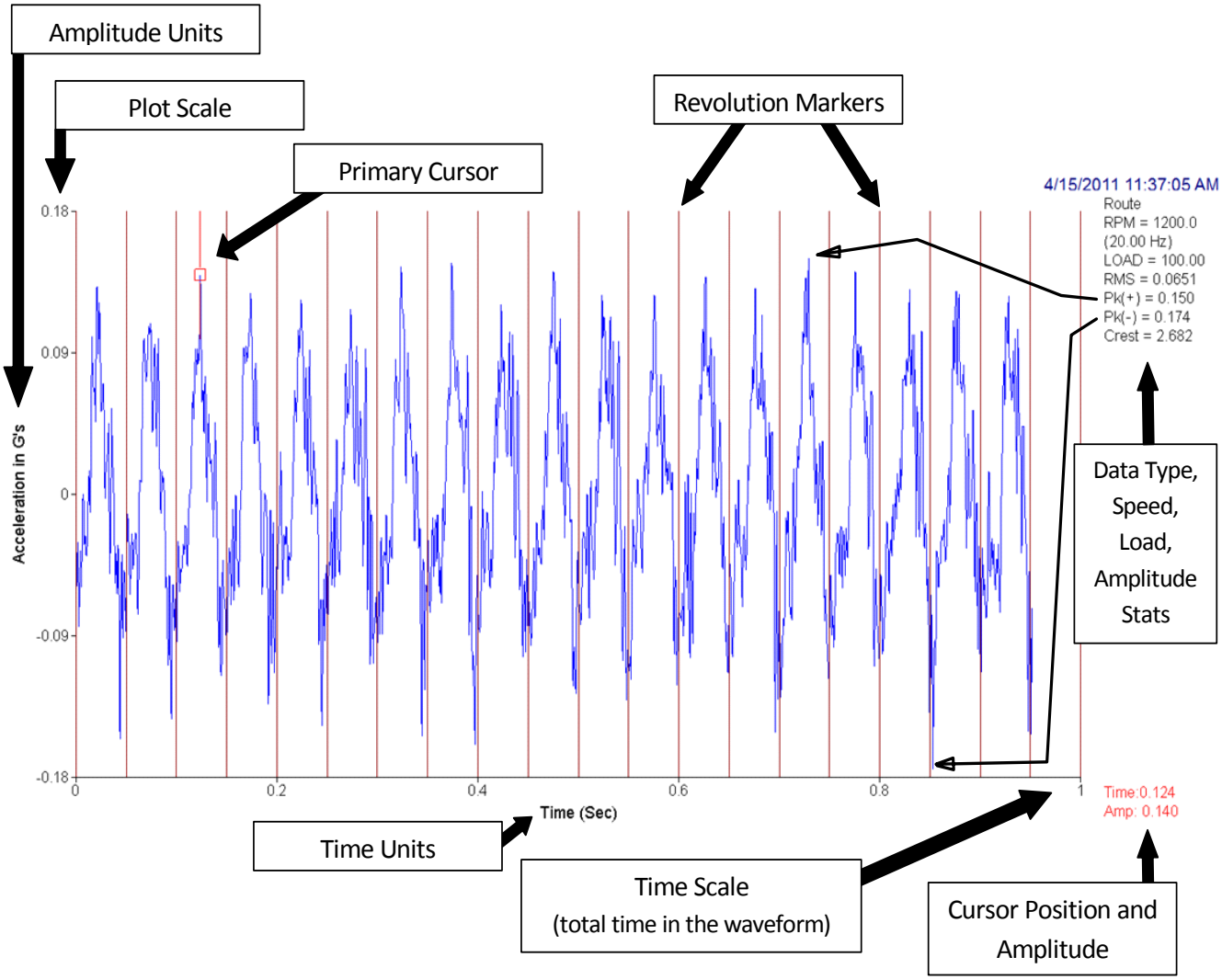
Waveform Data Display – Time Domain

Basic Elements of the Waveform Display



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In the sample Time Waveform:

- The **amplitude units** are in acceleration (G's).
- The amplitude scale in this example is **-** 0.18 G's to **+** 0.18 G's.
- The **time units** are in seconds. The **scale** indicates a total time of one second.
- The **Cursor** is placed on a peak marking the elapsed time from the starting point of the measurement up to .124 seconds into the measurement.
- The amplitude of the peak is .14 G's, as seen in the **cursor position and amplitude display** at the lower right hand corner.
- The **Revolution Markers** divide the waveform into time segments equal to the time it takes for the shaft to make one complete rotation. The accuracy of rotation markers are dependent upon the RPM entered by the user.
- The **Date and Time** of the measurement are displayed in the top right hand corner. The data was collected on 4/15/2011 at 11:37 AM.
- The data type is **Route** data as indicated just below the data and time in the top right hand corner.
- The **RPM** is 1200 (20 Hz) **-** entered by the analyst either during database set up or at the time of data collection.
- The **LOAD** is 100. The default will remain at 100 unless this parameter is entered manually at the time of data collection for trending purposes.
- The **RMS** (Root Mean Square) Value is .0651. This is the average RMS value of the entire waveform.
- The **Peak Positive** is at 0.150 **G's** and the **Peak Negative** is at 0.174 **G's**. This is known as the **Peak to Peak** waveform value, representing the maximum positive peak amplitude and the maximum negative peak amplitude present in the waveform. This is also known as a G-swing. In this example, the G-swing has a peak to peak value of 0.324 G's. This parameter value can be trended for analysis purposes.
- The **Crest Factor** is 2.682. This is the ratio of the Peak absolute value to the RMS value. Here, the highest peak value of .174 is divided by the RMS value of .0651 to get the Crest Factor. A high crest factor indicates that the individual peak values are well above the average peak values.

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Each of the listed items will play a role in the analysis process. Among these, the **amplitude units** and the acceleration **Peak to Peak values** will be the initial focus for the analyst. The amplitude reflects the severity of the vibration.

The time waveform contains very important information. Although many analysts do not view the time waveform during data analysis, it should be noted that the **waveform data is the source** that is used to generate the Spectral and Trend data that are commonly used for fault detection.

The cursor can be used to identify amplitude levels at a specific time within the measurement. The cursor can also be used mark peaks and calculate frequencies of individual vibration cycles.

The length of the waveform along with the amount of shaft revolutions is also important. The time waveform should be long enough to include 10 to 15 shaft rotations. The length, or the total amount of time in the waveform, is calculated by dividing the lines of resolution by the Fmax.

Understanding the relationship between the waveform and the spectrum is an important step to becoming a successful analyst.

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Time Waveform – Characteristics and Terminology

Two terms commonly used to describe vibration are **random** and **periodic**. Viewing the time waveform data can help identify and differentiate between these two types of vibration.

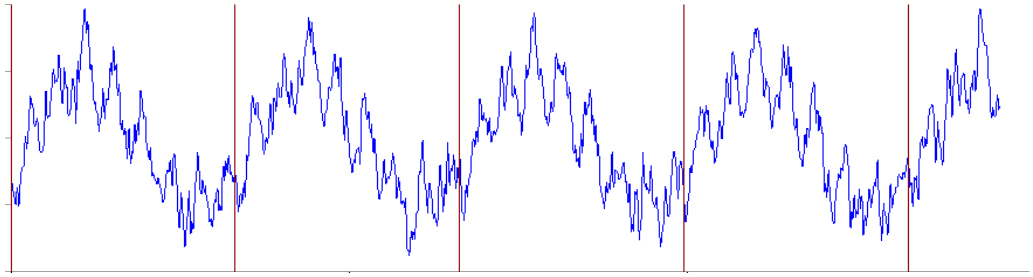
Periodic events in the waveform appear as repetitive cycles of vibration. Periodic vibrations may occur at any frequency, yet in a distinctive repetitive manner. These periodic cycles are often easily recognized by the analyst.

Random vibrations or occurrences in the waveform will appear as events or cycles that do not have a repetitive pattern. A random event may appear as little as once or twice in the stored waveform.

Different terms are used to describe the shape or characteristics of the waveform based on how the machine is vibrating. It is common for the waveform to have a combination of these characteristics depending on the type of fault.

Common Terms

- **Sinusoidal** – a waveform that resembles the shape of a pure sine wave in pattern with little to no impacting. Usually, one noticeable event per shaft rotation. The sample below shows one clear event per rotation with about eight smaller periodic events per rotation.



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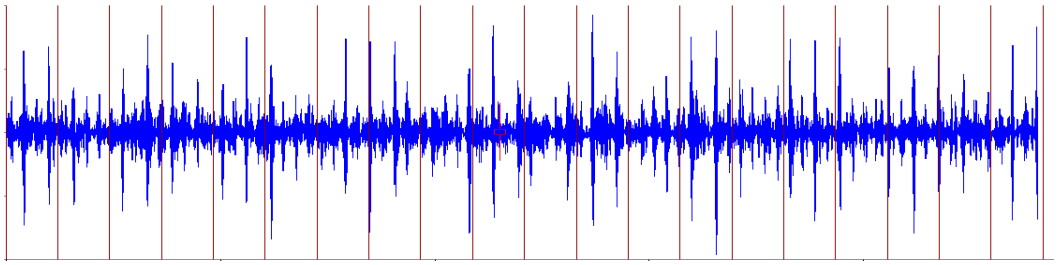
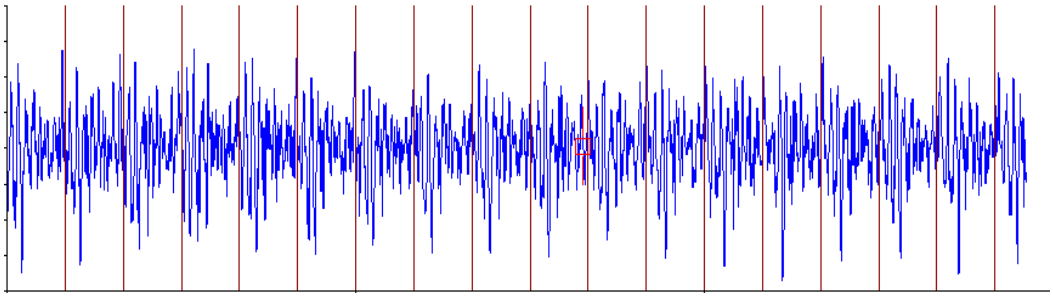
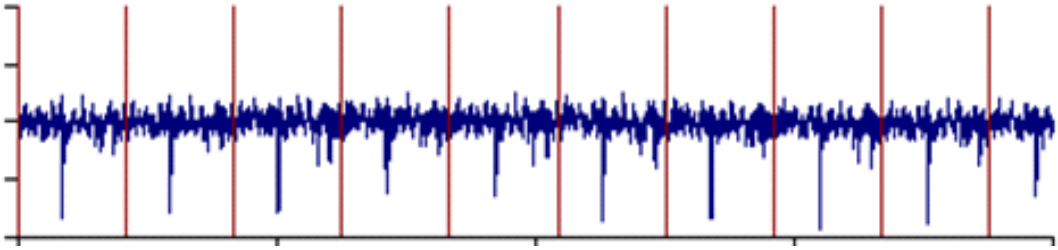
- **Impacting** – distinctive sharp peaks or spikes with amplitudes well above the average RMS value. Many times, a waveform with high levels of impacting will have a high crest factor. The impacting in this waveform example appears to be periodic. This is characteristic of several different types of mechanical defects. Impacting can be very low or very high in amplitude. Impacting in the waveform typically indicates a problem in the machine.



- **Asymmetrical** – a waveform with a lack of symmetry between the positive and negative values. In the example below, the amplitudes are greater on the bottom half of the waveform than in the top half. This is an indication that the machine or bearing had a higher degree of freedom to move in one direction rather than the other during the measurement.



- **Truncation** - typically asymmetrical in shape with short peaks that are "cut off" (truncated) on one side and higher amplitude spikes on the other. This is characteristic of a machine that has a rub or possibly a severely loose condition where motion is restricted in one direction by an impact.



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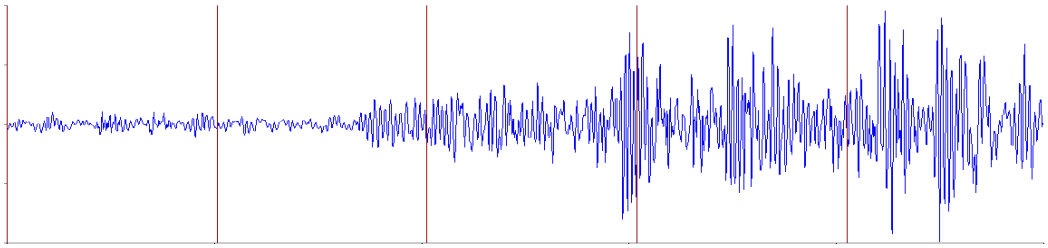
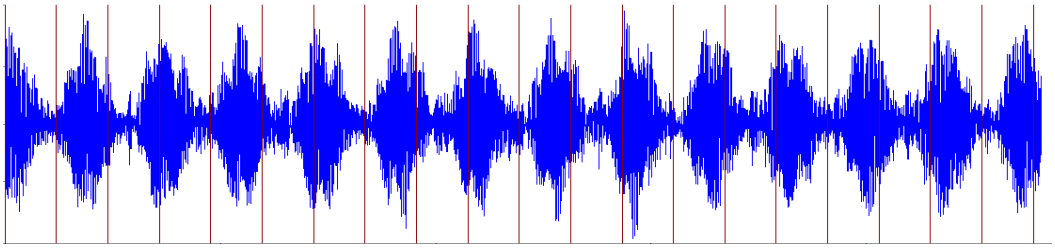
- **Transient or Random** – a waveform that is predominantly non-periodic, with events occurring only once or twice in a non-repetitive manner. Many times these events are present as impacts throughout the waveform, yet still very random in nature.



- **Modulation** – a measurable increase and decrease in amplitude or frequency levels in the waveform. Sometimes clearly evident, but not always. The waveform below is an example of amplitude modulation and is very repetitive.



The terms and characteristics used to describe the different waveform shapes is not limited to those listed here. However, these terms will be used when describing specific fault characteristics in the Machinery Faults section of this course manual.

Depending on the condition of the machine being monitored, the waveform could contain any combination of these characteristics. Recognizing these and other waveform characteristics is an important step for the analyst when determining the fault type and severity of the problem.

For mechanical faults that cause impacting, the true amplitude of the vibration can only be seen in the waveform. The primary reason for viewing the waveform is to help determine the severity of a problem once a fault has been identified.

The waveform may be viewed in any of the three amplitude units; displacement, velocity or acceleration. It may be necessary to change the amplitude units when looking for specific patterns or fault characteristics. Impacting in the waveform will be more evident when viewing the waveform in units of acceleration.

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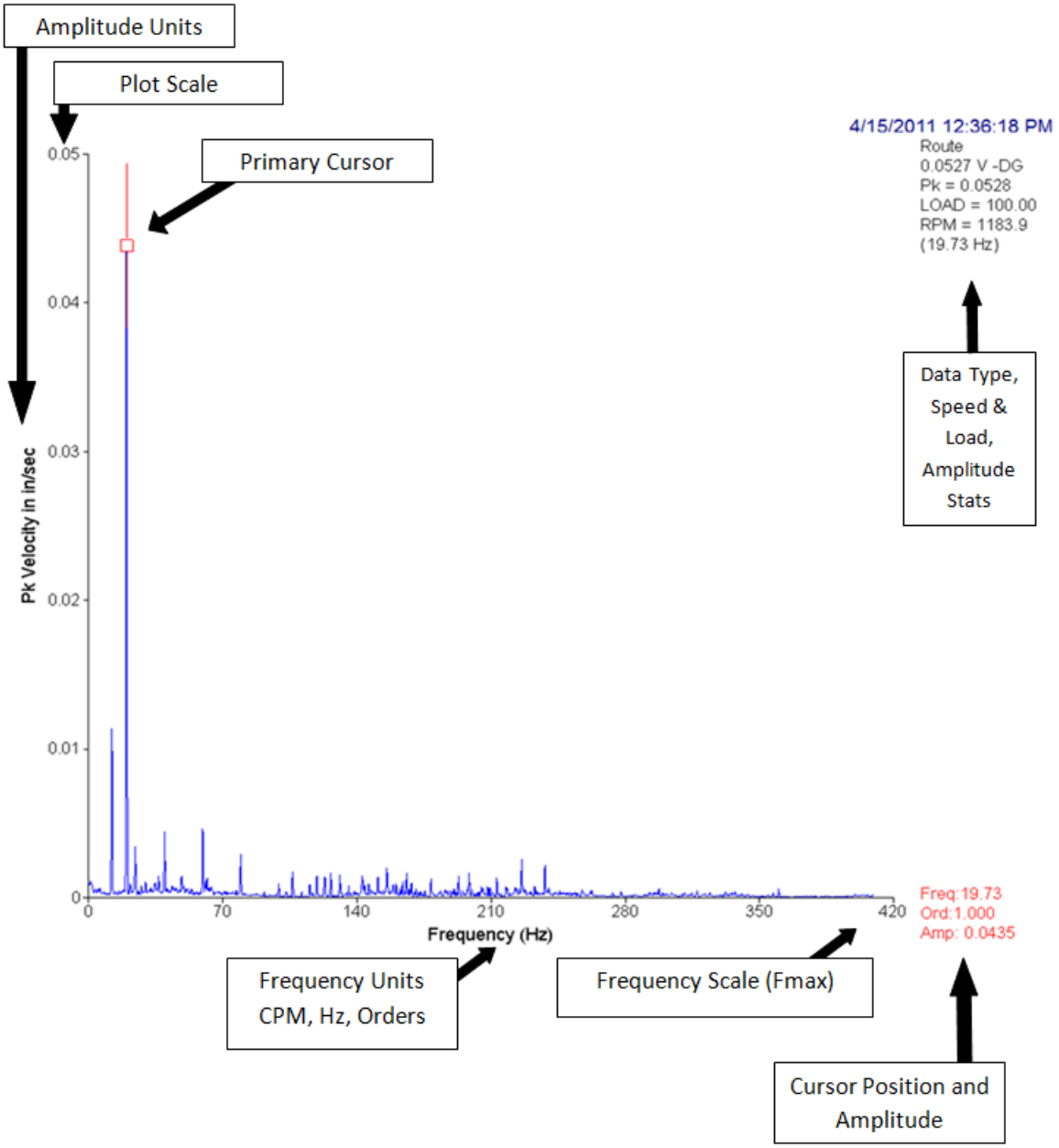
Spectrum Data Display – Frequency Domain

Basic Elements of the Spectrum Display



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In the sample Spectrum:

- The **Amplitude** is in velocity on a scale of 0.0 - 0.05 in/sec.
- The **Frequency** is in Hz. Other options are CPM and Orders
- The **Frequency Scale** indicates a maximum frequency of 420 Hz.
- The **Primary Cursor** is placed on a vibration peak to mark the frequency and the amplitude. The amplitude of the peak is .0435 in/sec, and the frequency is 19.73 Hz, as seen in the **cursor position and amplitude display** at the lower right hand corner. The relationship of the 19.73 Hz peak to the turning speed is displayed as 1 Order.
- The **Date and Time** of the measurement are displayed in the top right hand corner. The data was collected on 4/15/2011 at 12:36 AM.
- The data type is **Route** data as indicated just below the data and time in the top right hand corner.
- The **Overall vibration** is listed as 0.0527 V-DG. The **V** indicates velocity units. The **DG** indicates that the overall energy has been calculated digitally - from the spectrum.
- The **Pk** value in the viewed spectrum is .0528. The zoom feature will update this peak value based upon the frequency scale.
- The **LOAD** is 100. The default will remain at 100 unless this parameter value is entered manually for trending purposes. The default load may be set at the measurement point set up.
- The **RPM** is 1183.9 (19.73 Hz). The RPM value is presented in frequency units of Hz, since the frequency units for the spectrum are in Hz. The RPM is entered by the analyst either during database set up or at the time of data collection.

When viewing the spectrum display, the analyst should initially observe the **amplitude units**, the **plot scale** and the **overall value**. The overall value at the top right hand corner is trended by default, and is also available when viewing trend parameters for each measurement.

The spectrum may be viewed in any of the three amplitude units when required. The most common spectral amplitude unit for most route based measurements is velocity, due to velocity treating amplitude equally across the spectrum.

Since the data in the spectrum originates from the waveform, peak patterns that are present in the spectrum can be attributed to the vibration characteristics in the waveform.

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Spectral Data – Characteristics and Terminology

Common Terms and Tools Used with Spectrum Analysis

- **Frequency Patterns -** certain machinery fault conditions can cause different recognizable patterns in the spectrum. Pattern recognition is frequently used to identify some common defects such as imbalance, misalignment, and looseness.
- **Calculated Frequencies –** vibration frequencies that can be calculated from the known geometry of components within the machine. There are two categories of calculated frequencies.

  Normal – vibration peaks that appear in the spectrum as a normal condition of the function of the machine such as running speed, gear mesh, pump vane pass, etc.

  Defect (Fault) – calculated frequencies that appear in the spectrum only when there is a problem with the component such as a bearing defect frequency.
- **Fundamental Frequency** – usually a pre-calculated frequency, such as turning speed, or a frequency that is equal to a specific component vibration or defect in the machine.
- **Dominant Peak** – a peak in the spectrum with the highest amplitude value.
- **Harmonics –** integer multiples of any frequency. Harmonics in the spectrum are caused by harmonic distortion in the waveform.
- **Harmonic Family** – a group of harmonics and/or harmonically related peaks.
- **Harmonic Cursor –** used to identify harmonic peaks of a fundamental frequency.
- **Sidebands** - a pattern of equally spaced peaks above and below a center calculated frequency. Sidebands are caused by modulation of a carrier frequency.
- **Carrier (center) Frequency –** a vibration peak that is either changing in amplitude or frequency. The carrier frequency is the “center” frequency that appears between the sideband peaks.
- **Sideband Cursor** – used to identify sideband peaks of a center frequency.
- **Reference Cursor –** marks a peak to compare the difference from the primary cursor position.

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- **Broadband Energy** – an area in the spectrum where peaks are so close together they appear as a single mound of energy. Often called a raised noise floor.
- **Order** – identifies the relationship between the frequency of a peak and the rotating speed of the shaft. Considered as a frequency unit – but **only when shaft turning speed is known**.

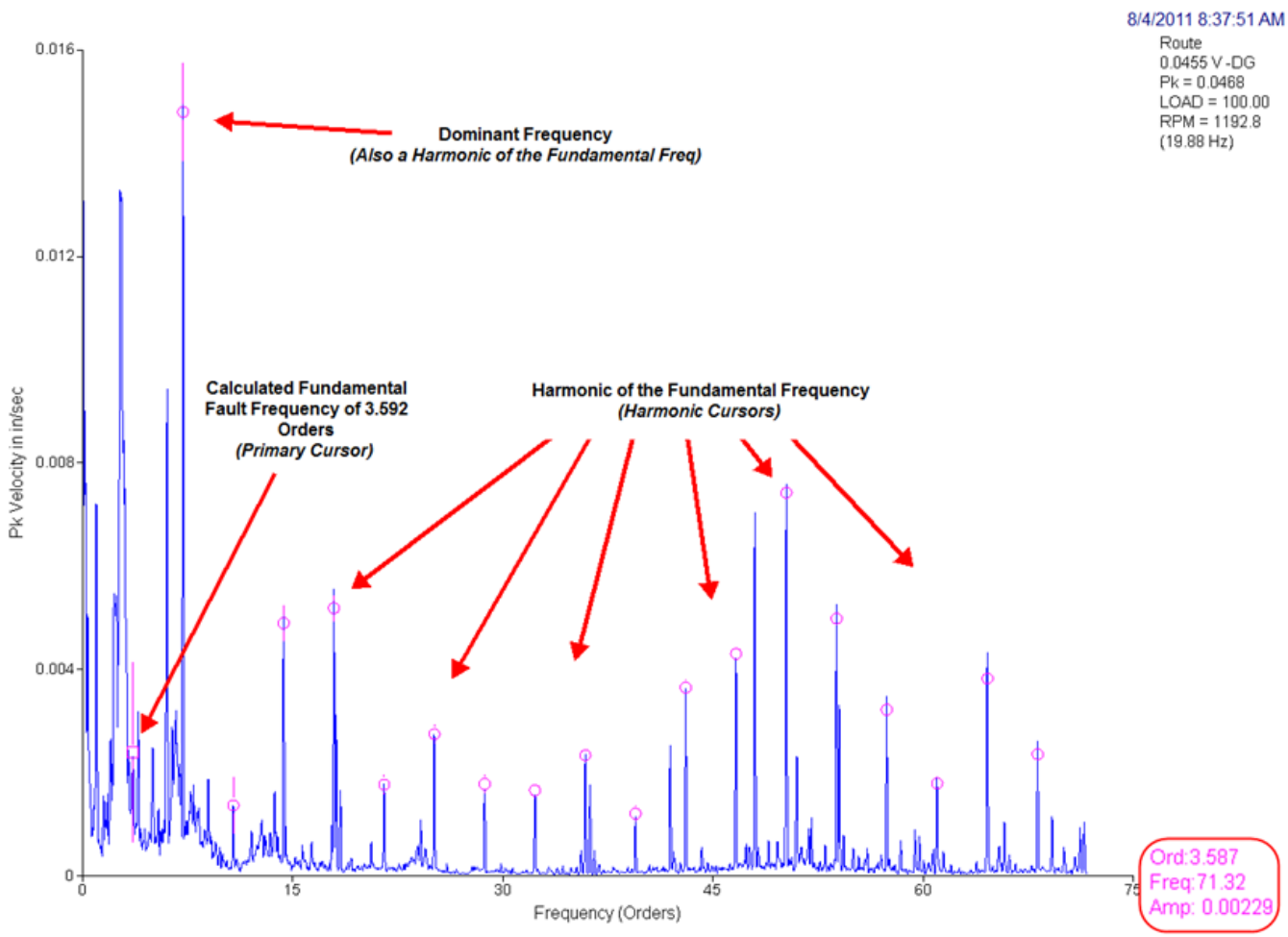
Harmonics

Harmonics are integer multiples of any frequency;

(1 x *f*, 2 x *f*, 3 x *f*, etc.)

$$\boldsymbol{f} = \mathbf{N} \times \boldsymbol{f}$$

Where ***f*** is a given frequency, and **N** is some whole number (1,2,3,4, etc.)



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In this spectrum, the **fundamental frequency** is marked and labeled at 71.32 Hz. The harmonically related peaks are labeled at integer multiples of 71.32 Hz starting at 142.64 Hz, 213.96 Hz, etc. This is an example of a calculated bearing defect frequency with **harmonics**.

If harmonic peaks are present in the spectrum, this is usually an indication of some level of impacting in the waveform. Impacting in the equipment is not common for most industrial equipment and usually indicates a potential problem. In this example, the waveform would contain impacting at a frequency of 71.32 Hz.

Peaks that are harmonically related are caused by the same defect. The spacing of the harmonics will be equal to the fundamental frequency that is causing the harmonics, and therefore represents the source of the impacting. Use the harmonic cursor to identify harmonically related peaks in the spectrum.

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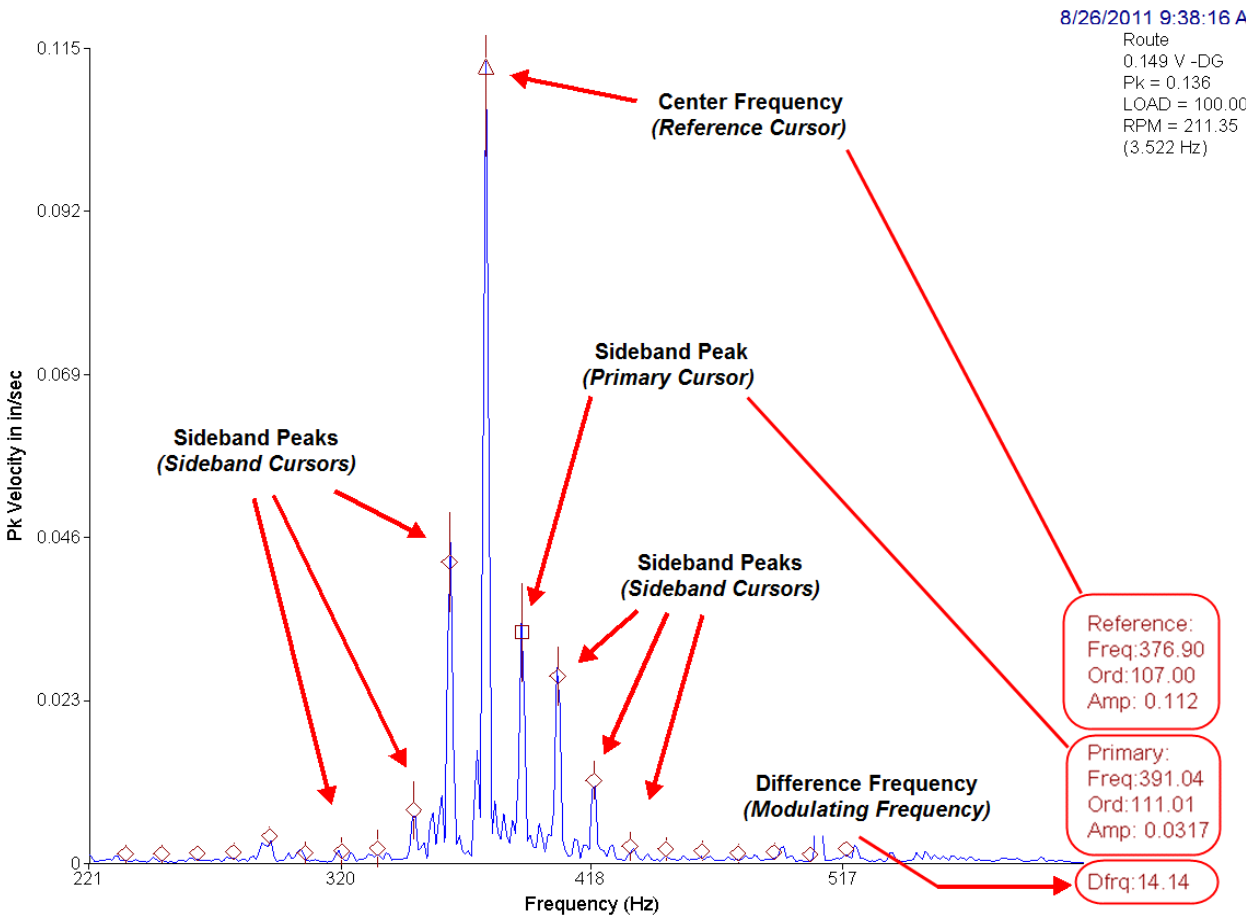
Sidebands

Sidebands in the spectrum are equally spaced peaks above and/or below a **center frequency**.

Sidebands are caused by a modulating frequency in the waveform.

This center frequency is called the **carrier frequency**. In the waveform, the carrier frequency is modulating (usually changing in amplitude). When the FFT process is applied to the waveform, the carrier frequency appears as a single peak in the spectrum along with one or more sideband peaks. This group of peaks is sometimes called a sideband family of peaks.



The Dfrq (difference frequency) is the spacing between the center frequency and the sideband frequencies indicating the frequency at which the modulation is occurring. Many times this modulating frequency leads to the potential source of the problem.

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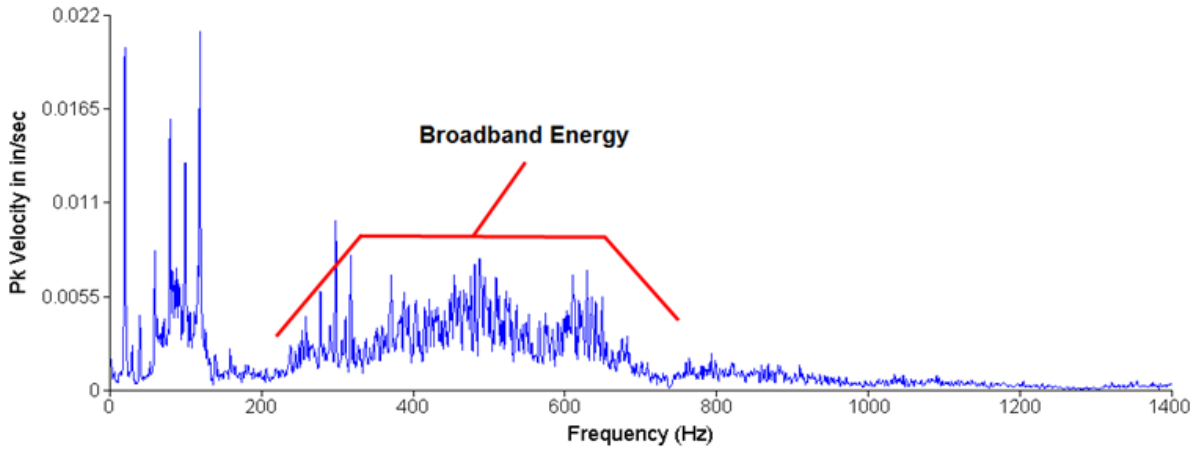
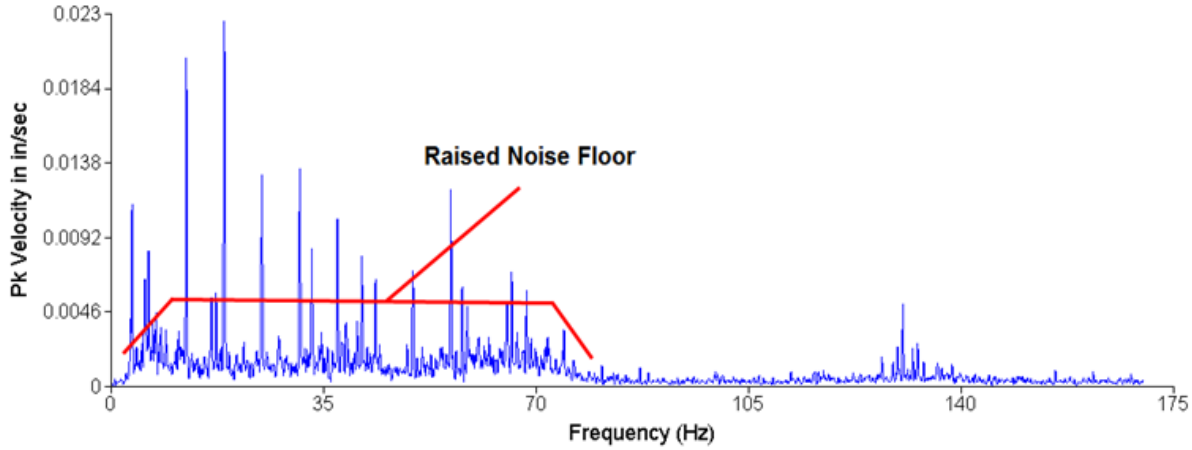
Broadband Energy

Broadband Energy appears in the spectrum as a mound of peaks with a raised noise floor. Broad banding is caused by random, non periodic impacting in the waveform.

Broad banding usually appears in a localized area of the spectrum, depending on the type of problem, as seen in the example below.



Broad banding may also appear as a raised noise floor across a wide range of peaks, as seen below.



Locating the source of a forcing function requires the analyst to first understand how the peaks in the spectrum relate to the components in the system being monitored.

If the turning speed of the rotating component is known at the time the measurement was taken, the relationship between any frequency and the shaft speed can be displayed accurately. This information is very essential to identifying the source of the forcing function and how it relates to the machine components.

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Reminder - the relationship of any frequency to the rotating frequency is called an **order**.

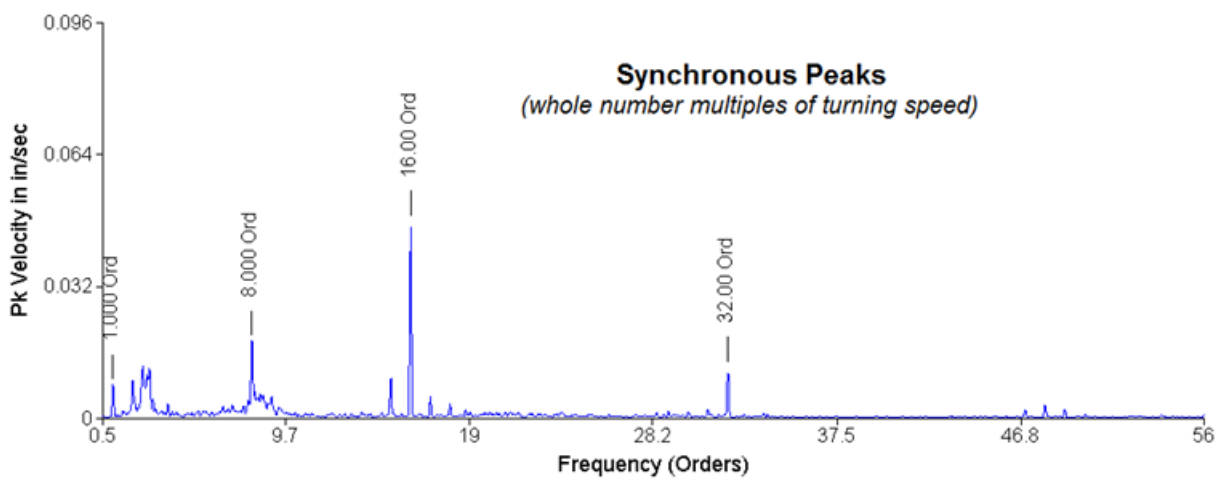
To simplify the analysis process, the energy in the spectrum is divided into three categories; **Synchronous, Nonsynchronous, and Subsynchronous**.

Synchronous

Spectral peaks at 1 x TS, or as a whole number multiple of 1 x TS.

Possible causes of **Synchronous** vibration peaks in the spectrum include:

- Imbalance (1x turning speed only)
- Pitch-line run out (PLRO) in gears and belt sheaves
- Bent shaft
- Misalignment
- Looseness
- Blade pass frequency in blowers/fans
- Vane pass frequency in pumps
- Reciprocating motion
- Gear mesh frequency
- Stator slot frequency of motors
- Rotor bar frequency of motors



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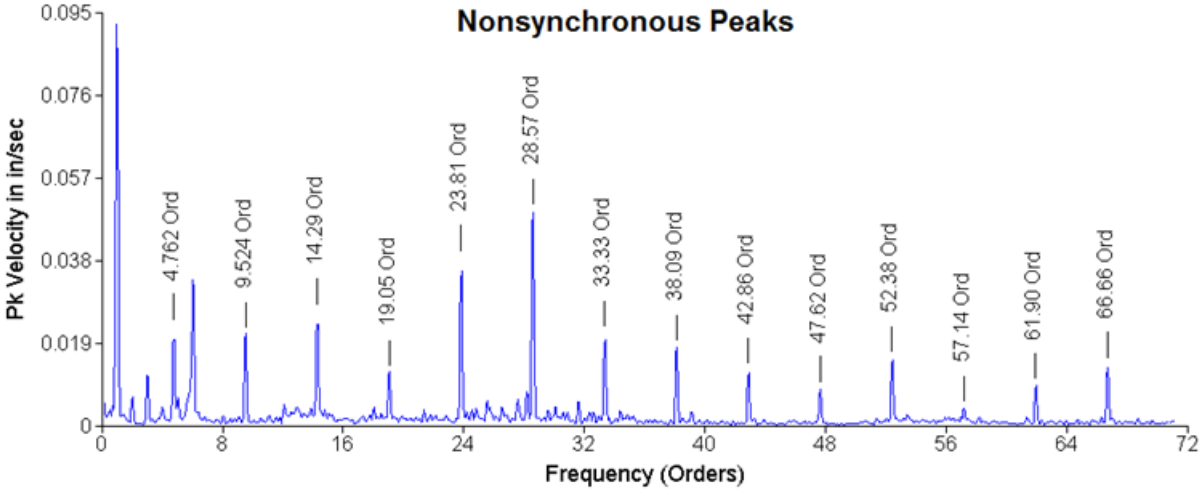
Nonsynchronous

Spectral peaks that are above turning speed (1 x TS), but are not whole number multiples of turning speed.

Possible causes of **Nonsynchronous** peaks in the spectrum are:

- Vibration from another component or machine
- Higher multiples of belt frequency
- Rolling element bearing defects
- System resonances
- Electrical faults
- Centrifugal clutches
- Compressor surge



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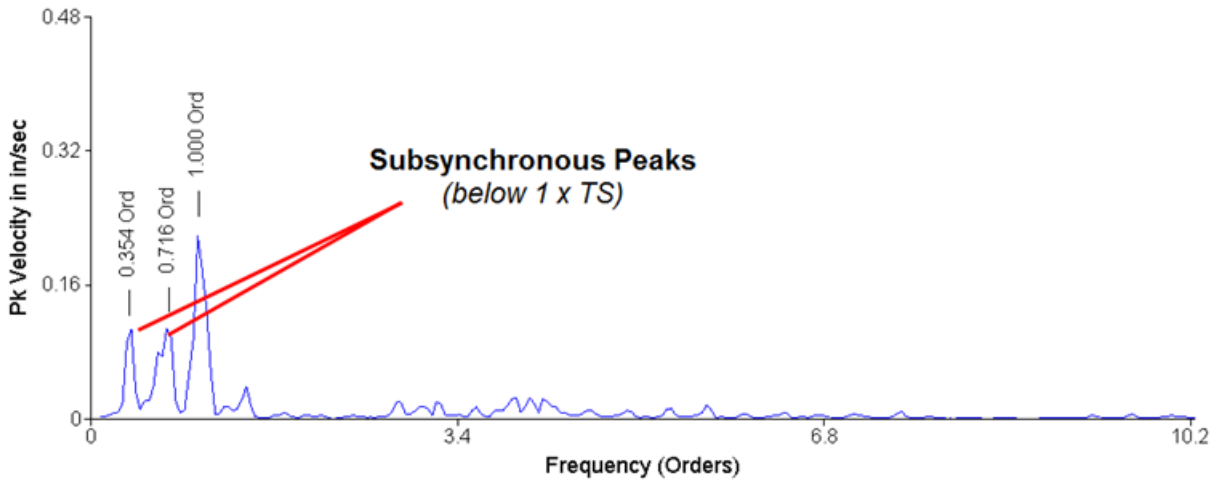
Subsynchronous

Vibration at frequencies lower than 1 x TS.

Possible causes of **Subsynchronous** peaks in the spectrum are:

- Vibration from another component in the machine that rotates at a slower speed.
- Vibration from another machine in the area.
- Drive belt defects.
- Hydraulic instability such as oil whirl or oil whip
- Rotor or shaft rub
- Cage frequency of antifriction bearings



Note

Important – The turning speed of the shaft must be set accurately in the spectrum before the analyst can rely on the three categories of energy described in this section.

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Detecting a Machine Fault or Problem in the Data

Fault analysis should start in the field during the data collection process, beginning with an assessment of the overall value and how it is trending.

By default, most vibration data collection systems acquire and trend the overall energy level for each measurement. The overall value is typically derived from the spectrum using the spectral parameter settings, but may also be calculated from the overall energy in the waveform.

The overall value may be useful in determining the condition of the machine. However, since the overall only **represents a measurement** of the total energy in the user defined frequency range, it gives very little indication as to whether there is a mechanical problem or where the source of the problem resides.

Therefore, it is suggested that the overall value be used as a reference, not for determining the source of the problem.

During data collection, after a measurement has been stored in the analyzer, the Overall vibration level should appear on the screen. Check the overall value, comparing it to the levels from the previous survey. Also, view the individual parameters and the status information for each parameter.

The status will be one of the following:

- **OK** – No parameter values are in alarm, no further analysis needed.
- **Warning** – The Baseline ratio (Br) alarm has been exceeded by one or more parameters.
- **High Alarm 1** – The Overall value has exceeded the Alert level as defined in the database setup.
- **High Alarm 2** – The Overall value has exceeded the Fault level as defined in the database setup.
- **Vib Alarm** – The Alarm level has been exceeded by one or more parameters.

Once it has been decided that there is indeed a problem, the next step towards finding the source and severity of the problem is to perform spectrum and waveform analysis.

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Spectral Fault Analysis – A Basic Approach

There are some basic steps to performing vibration analysis. Not every problem can be diagnosed through basic spectrum analysis, but by following a few simple steps the problem can usually be narrowed down to one or two possible causes.

For the most part, the frequency of the vibration indicates the cause or nature of the problem. The amplitude indicates the severity.

The fault analysis process may be approached using several different methods based on experience, knowledge of the machine, type of fault suspected, etc.

Some options for data analysis are:

- The process could begin by viewing the spectral data right away, skipping trend and waveform data, viewing spectra from each measurement point looking for specific faults in the data until all data points have been viewed.
- Another approach may be to begin by viewing the trended values of different parameters, then drill down to access the spectrum and waveform plots when a trend value is considered to be high or has reached an alarm.
- A third approach may use the exceptions reporting tools to list or display only the measurement points that have exceeded an alert or alarm, thus limiting the analysis process to the machines requiring attention. This approach would require that the alarm levels have been set properly and are reliable.
- Another alternative would be to view a trend of the Overall, a single spectrum, and a waveform of each measurement point collected.

In any case, when a problem is suspected, the process will eventually require the analyst to view the spectrum to allow for a detailed analysis of the peaks that represent the forcing functions in the system.

Once a fault has been identified in the spectrum, viewing the waveform will help determine the severity. Parameter trend values may eventually play a role in the final diagnosis as well.

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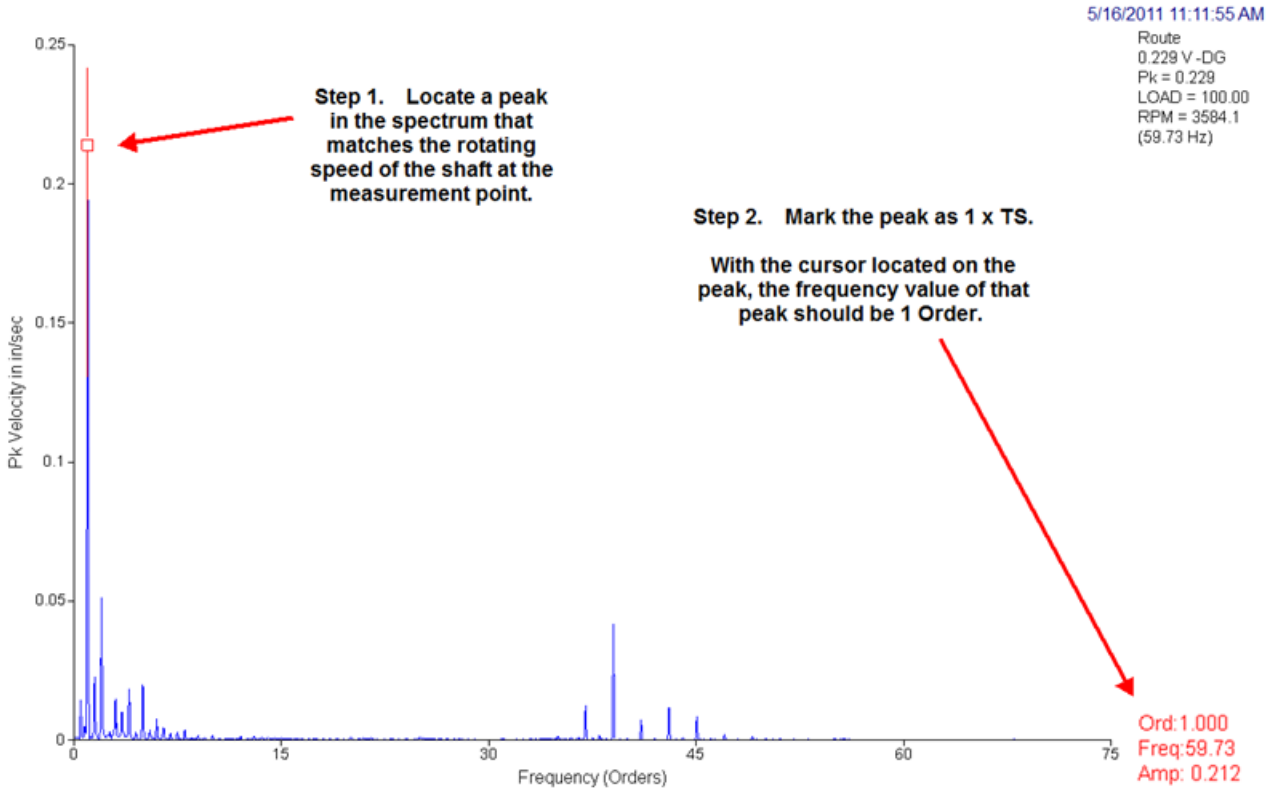
Recommended Steps for Basic Spectral Analysis

1. Locate the peak in the spectrum that is equal to the operating frequency of the shaft or rotating component. The 1 x TS peak is not always easy to identify. The shaft speed must be known at the measurement point before the 1 x TS peak can accurately be located in the spectrum.

   Unless there is enough vibration at shaft turning speed during data collection to cause a peak in the spectrum, the analyst must rely on other means of identifying the 1 x TS peak, such as a strobe light.

2. With the 1 x TS peak marked, set the RPM in the spectrum to allow for detailed analysis.

   It is extremely important that the analyst be able to identify the RPM. Without identifying the correct RPM, spectrum analysis is difficult if not impossible. Keep in mind, there may not be a significant vibration peak at turning speed, but the turning speed frequency must still be entered to allow for accurate detailed analysis.



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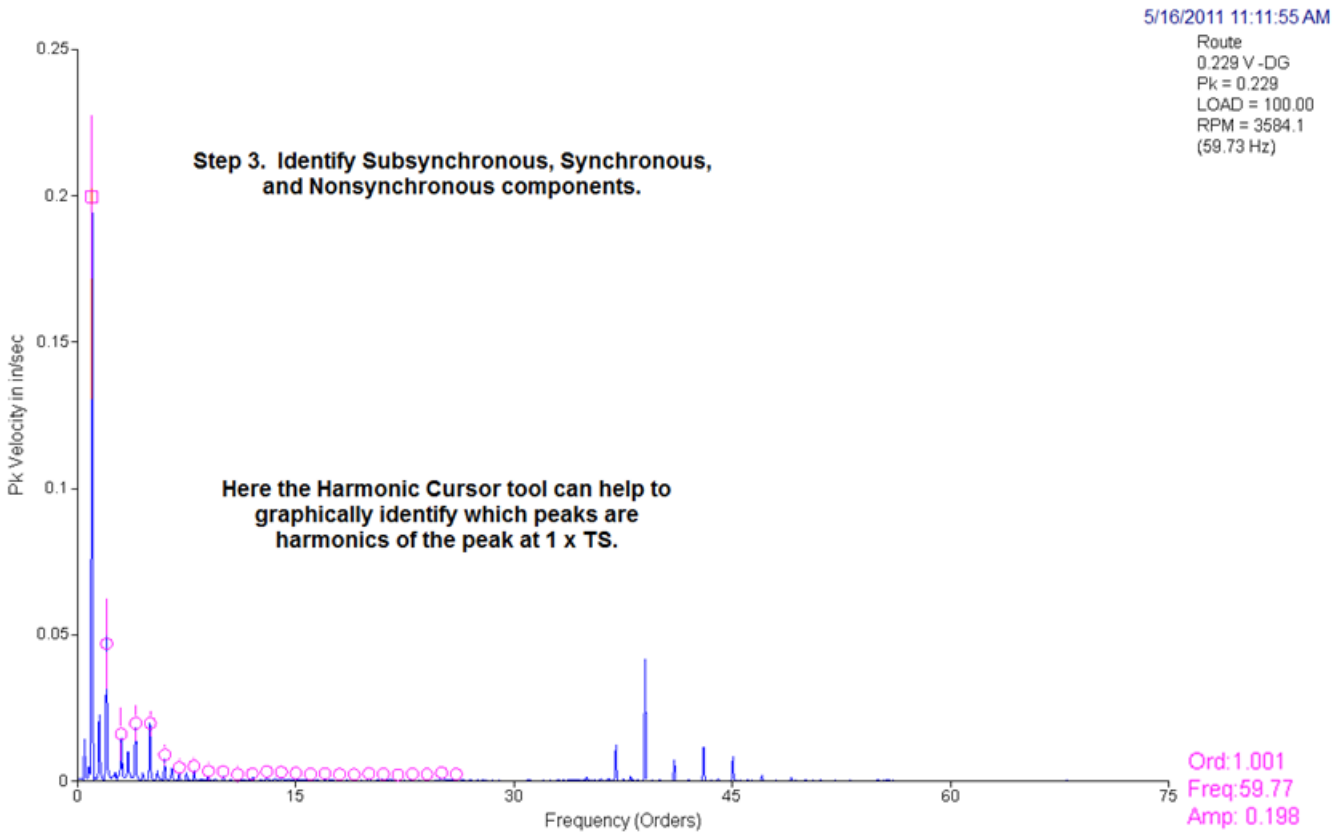
3. Determine how much energy falls into the following categories of energy:
   - Subsynchronous
   - Synchronous
   - Nonsynchronous

   Categorizing the peaks as Subsynchronous, Synchronous, and Nonsynchronous energy will help narrow down the list of potential forcing functions.

   The categorization process will help to eliminate certain faults. The AMS Machinery Manager software can generate a report indicating the percentages of each of these energy categories.



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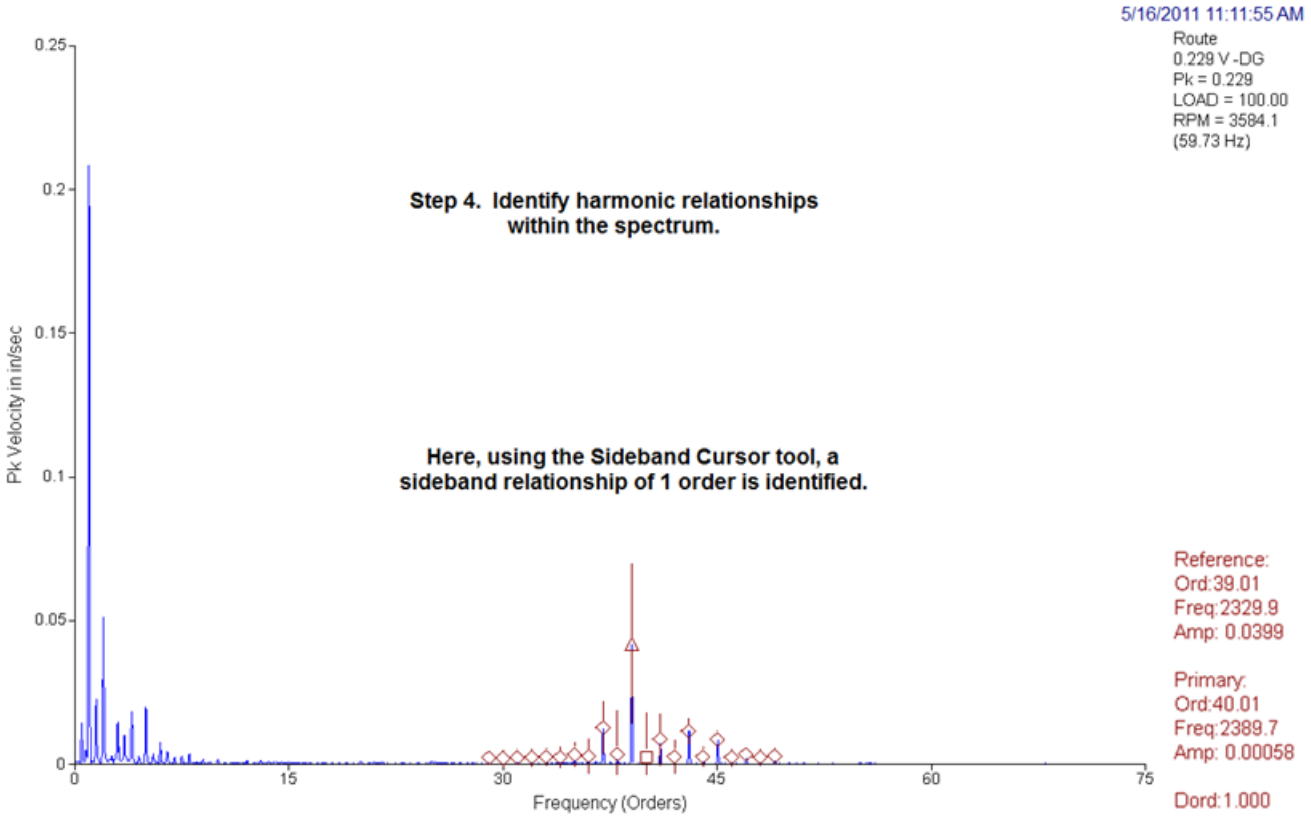
4. Identify harmonic relationships within the spectrum.

   Instead of identifying each discrete peak in the spectrum, it is often possible to identify harmonic families within the spectrum using the harmonic and sideband cursors. The significance of harmonics and sidebands is very useful in determining the potential source of a defect.

   Harmonics in the spectrum are usually an indication of some level of impacting in the waveform. The spacing of the harmonics represents the source of the forcing function causing the impact. There may be more than one harmonic family of peaks, depending on the number of forcing functions causing the impacting.

   Sidebands in the spectrum are an indication of modulation in the waveform. The spacing (difference frequency) between the center frequency and the sideband peaks will typically lead the analyst to the source or forcing function that is causing the modulation.



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5. Determine the severity.

   There are many factors to consider when determining the severity of a machinery defect. Machinery load, speed, and design will need to be considered, as well as the type of fault.

   When possible, use comparisons to help evaluate a machine's condition. Compare the levels with similar machinery with similar functionality and design. Comparing overall and peak levels to the levels from previous survey data can be very instrumental in successful fault and condition evaluation.

   Consider the amplitude scale on the left side of the spectrum and overall value of the energy displayed in the top right corner. Are these values higher than normal for this machine when compared to previous data or similar equipment?

   View the trend data of the measured parameters to determine if there is an increase near the frequency range of interest. Have the alarm values been exceeded?

   View the waveform amplitudes, looking for increases in impacting by viewing the peak to peak trend values. In addition to increases in amplitude values, many machine faults will produce significant pattern changes in the waveform. These pattern changes along with amplitude increases will help determine the severity.

Various organizations in industry have published standards of acceptable vibration levels for various types of equipment under various operating conditions.

Some of these organizations include ISO (International Standards Organization) and API (American Petroleum Institute). The equipment manufacturer may also have documentation about acceptable vibration levels.

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Section 1 Review

1. What is Vibration?

2. Two methods of acquiring vibration are to measure _____ movement and to measure_____ movement.

3. The word used to describe vibration as being repetitive is_____ .

4. Random vibration is the opposite of Periodic vibration. True or False

5. What is a waveform? Include explanation of the X and Y units of the waveform.

6. Name the three amplitude measurements and the common units of measure for each.

_____expressed in units of_____

_____expressed in units of_____

_____expressed in units of_____

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7. The amplitude of the vibration is helpful in determining the _____of the problem.

8. The mathematical process of converting Acceleration to Velocity and/or Velocity to Displacement is known as _____.

9. The time it takes to complete one cycle of vibration is known as _____.

10. Define Frequency.

11. How can a frequency be calculated from a time period?

12. The frequency domain is called a _____?

13. A spectrum represents _____ and _____.

14. The units of frequency are _____, _____ and _____.

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15. Describe the relationship between Hertz (Hz) and CPM.

16. Explain what is meant by the term Orders.

17. The process used to generate the spectrum from the waveform is called _____ _____.

18. What is the preferred amplitude unit when viewing the spectrum and why?

19. What is the preferred amplitude unit when viewing the waveform and why?

20. The spectrum is calculated using the values from the _____.

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21. Name at least four common terms used to describe waveform characteristics or shapes that may occur based on how a machine is vibrating.

    1) _____

    2) _____

    3) _____

    4) _____

22. What is modulation?

23. The waveform is typically viewed in units of _____ to allow the analyst to see evidence of impacting and the true amplitude of the vibration.

24. The spectrum is typically viewed in velocity. Why?

25. Calculated frequencies fall into two categories. Name the two categories and give an example of each.

26. What are harmonics in the spectrum and what causes them?

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27. What are sidebands in the spectrum and what causes them?

28. What is broadband energy in the spectrum?

29. Name the three categories of spectral energy used to simplify the analysis process when analyzing the spectrum.

    1) _____

    2) _____

    3) _____

30. True or False. The overall vibration level can help determine the source of a potential problem.

31. Typically, frequency helps determine the _____ of the vibration, while amplitude helps assess the _____ of the vibration.

32. Spectral analysis is difficult if not impossible without knowing the _____.

33. Harmonics in the spectrum are usually an indication of _____in the waveform.

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34. Sidebands in the spectrum are an indication of _____ in the waveform.

35. Name a few factors to consider when determining the severity of machinery defects.

36. Name three things that are helpful in determining the severity of a vibration problem.

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Digital Signal Processing

Section 2

Objectives

- Relate time waveform length and frequency bandwidth to sample rate and sample size.
- Determine the minimum resolution needed for specific applications.
- Recognize limitations of digital signal processing.

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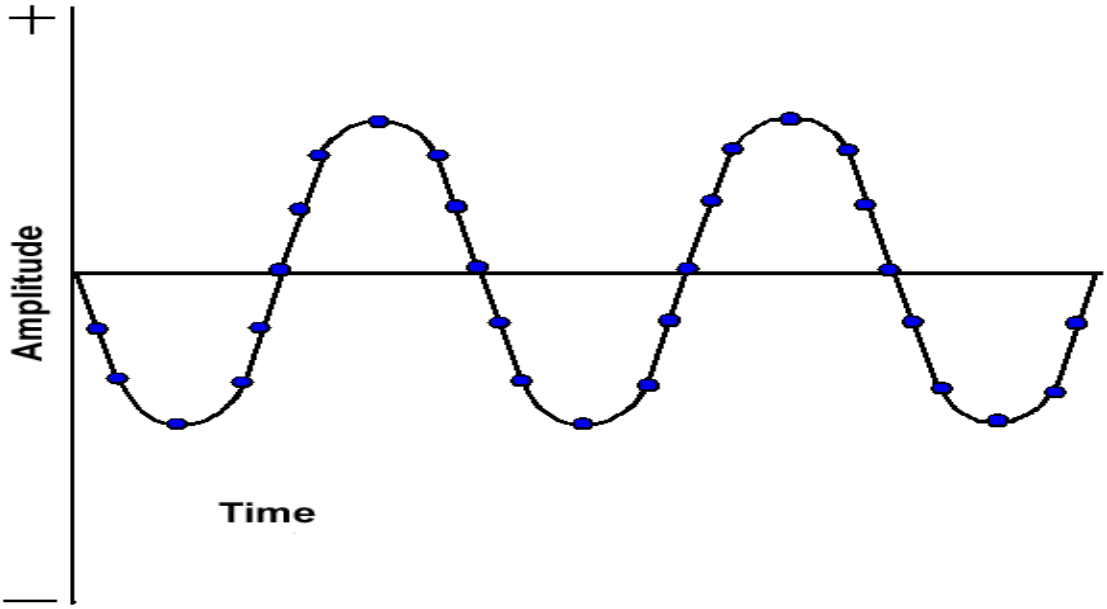
Time Waveform Capture

The modern data analyzer is a sampled data instrument and operates very differently from the analog oscilloscopes of years past. Those instruments displayed a continuous record of the incoming waveform for the viewer to see. Most were not capable of storing any information and had many other limitations. However, it was generally difficult to set the controls to get a totally correct image.

The sampled data analyzer only stores (and displays) a small piece of the data called a sample. The analyzer then waits one sample time and then repeats the capture and storage on the next sample.



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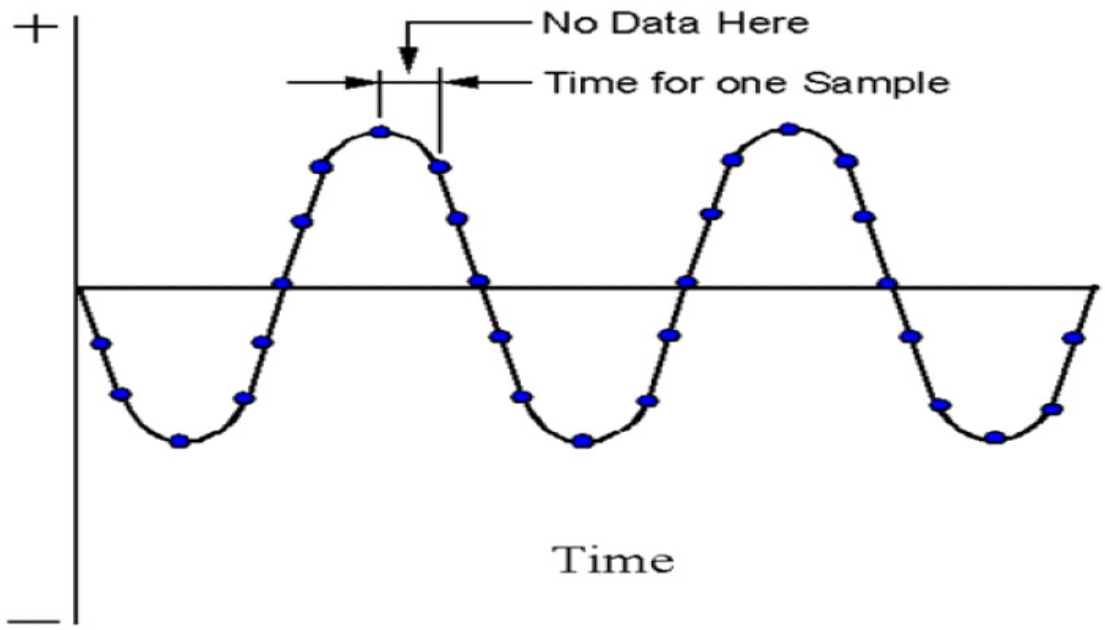
The time between the samples and the number of samples eventually stored are fully user controlled. Without a good understanding of the complexities and design of the instrument, it is possible to obtain results that do not represent the subject actually being analyzed. The drawing below illustrates the process.



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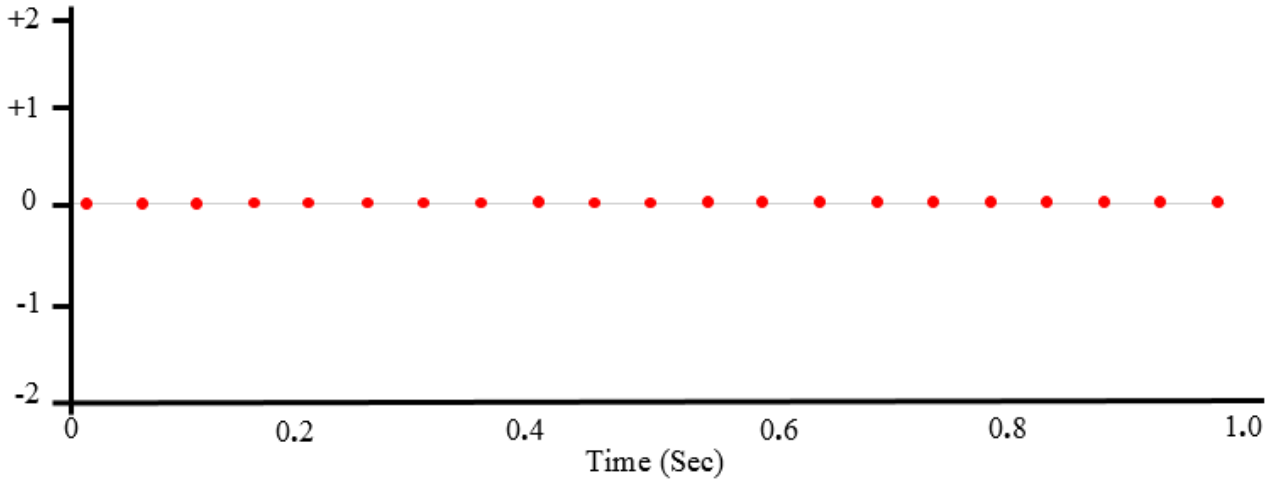
Aliasing

In a digital analyzer, the time domain data is digitally sampled at a controlled sample rate. This sample rate is controlled by the maximum analysis frequency (Fmax). In order to get good data, the sample rate must be set at least two times the maximum frequency of interest, Fmax. **Two times Fmax** is known as the **Nyquist frequency**.

Aliasing occurs when frequencies higher than the Fmax are present in the signal, and when no filter is set at Fmax to eliminate the higher frequency of vibration. The sample rate under samples this high frequency and creates a low (aliased) frequency equal to the high frequency minus sampling frequency.

To illustrate this idea, start with an analog waveform that has no signal. The sample rate is 20 Hz. In a one second waveform, we would have 20 samples. A new sample would be stored every .05 sec.



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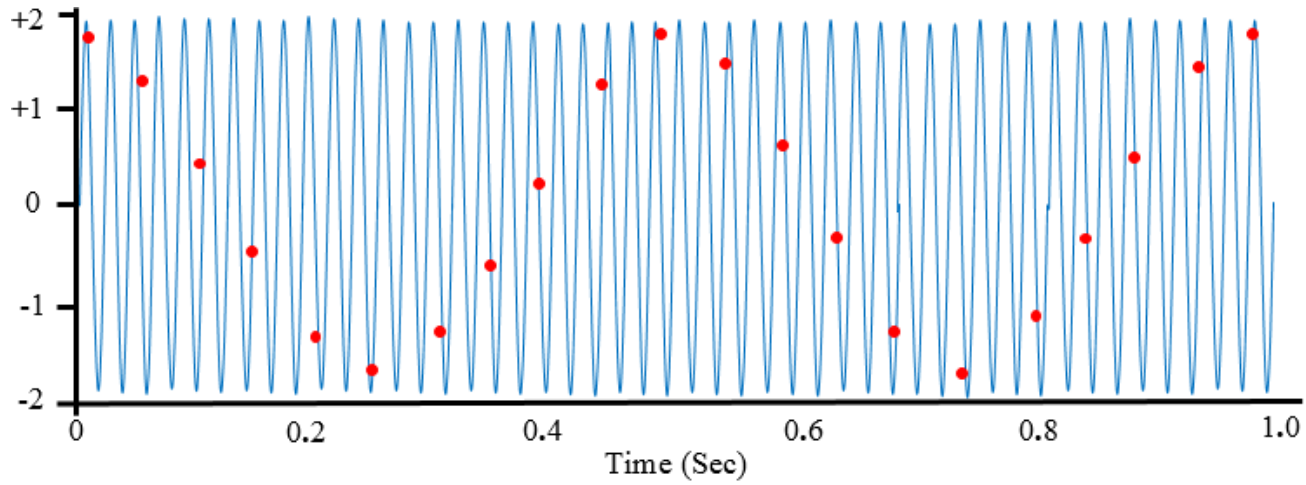
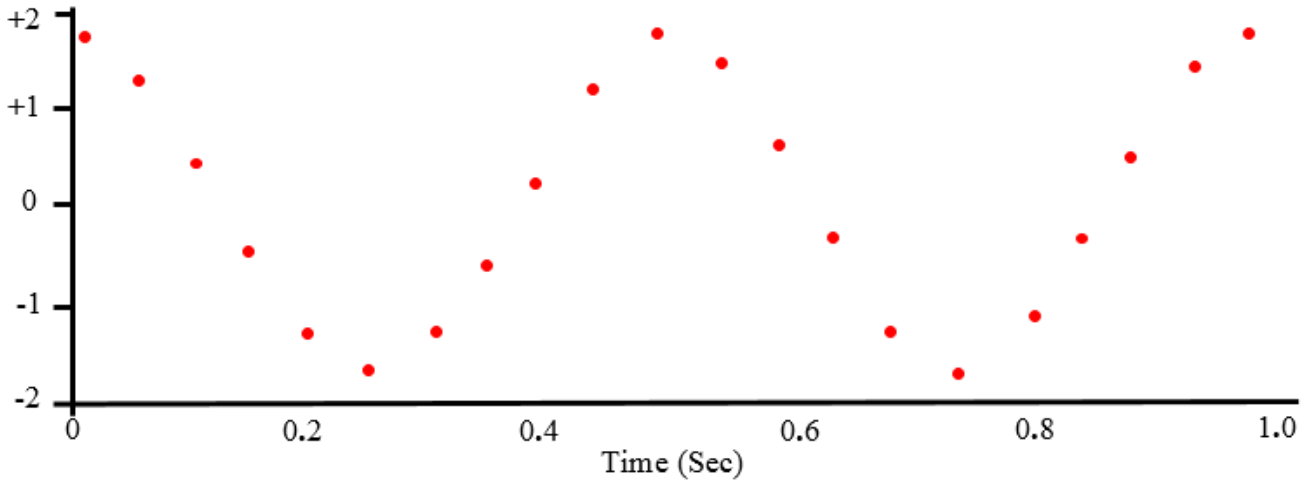
Introduce a vibratory signal of approximately 55Hz, for example. The vibration is at a much higher frequency than the sample rate. Remember, with digital sampling, nothing is known about the time in between the samples.



In the next step, the 55 Hz signal is removed. Only the digital samples are actually recorded.



There are now 20 digitally sampled points of data.

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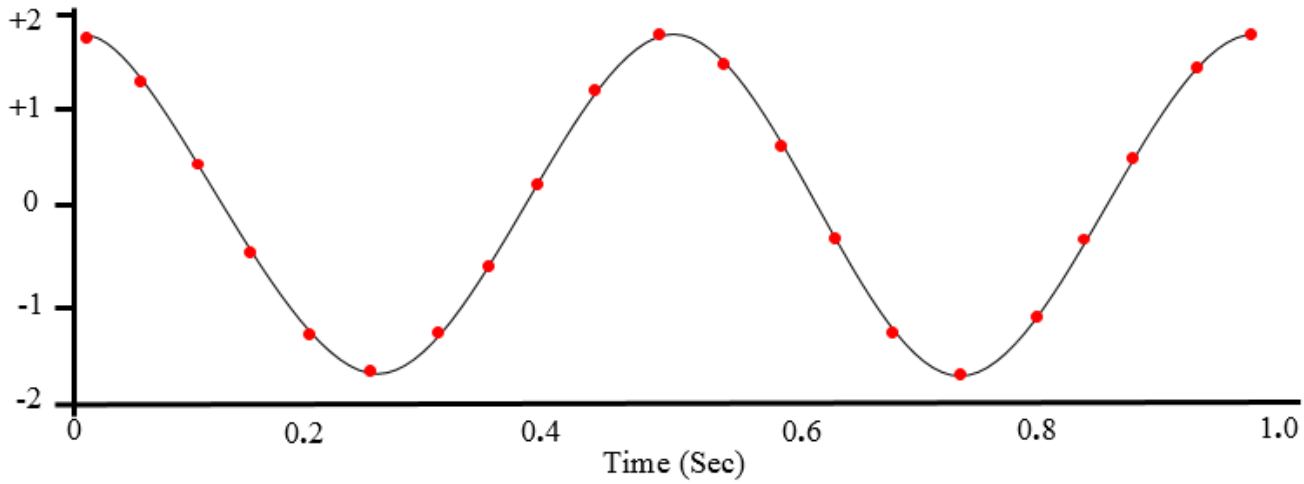
The next step is to connect the samples with lines, which creates the digital time waveform.



The problem is the digital waveform has a period of about .5 seconds or a frequency of around 2 Hz. The original analog signal was about 55 Hz. The 2 Hz signal is an aliased signal and does not actually exist. Due to the way the digital signal analyzer works, aliasing is not a problem.

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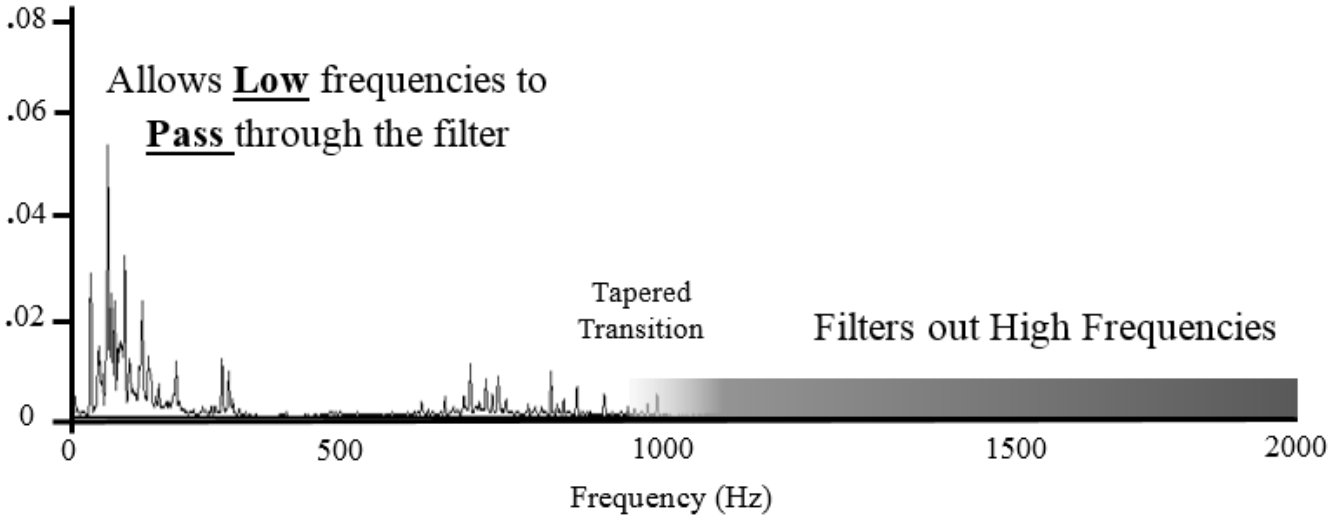
Two functions are performed by the digital signal analyzer to prevent aliasing.

First, the analog waveform is filtered to prevent higher frequency signals from affecting the data. The type of filter used in this case is referred to as a **Low Pass Filter**. This low pass filter is set to the spectral Fmax. For example, if the user sets a 1000 Hz Fmax, the analyzer applies a 1000 Hz Low Pass Filter to the waveform. Since this filter helps to prevent aliasing, it is sometimes referred to as an **Anti-aliasing Filter**.



Second, the filtered waveform is sampled at a rate that is higher than the low pass filter setting. According to the **Nyquist Sampling Theorem**, the highest frequency that can be accurately digitally sampled is a frequency exactly half that of the sampling rate. This means that in order for the digitally sampled data to be accurate, the sampling rate needs to be at least twice what the highest vibration frequency is. When acquiring normal vibration data, Emerson digital analyzers always use a sampling rate that is 2.56 times faster than the Fmax.

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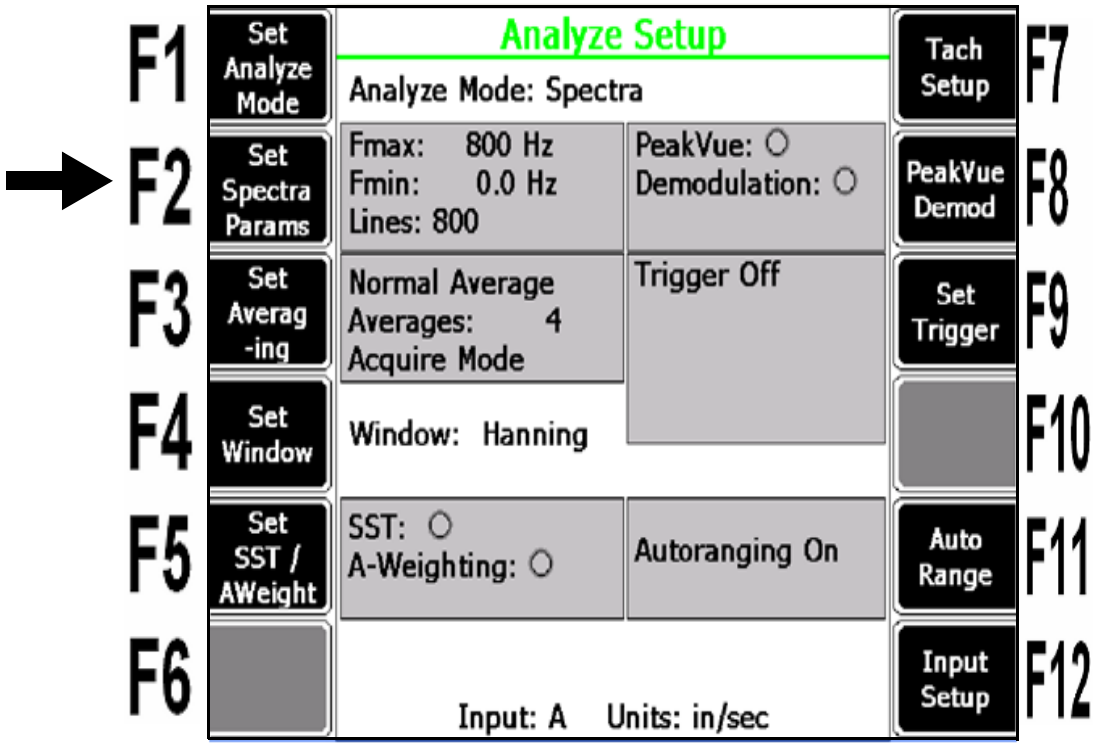
Spectrum/Waveform Setup Exercise

Consider the typical spectrum setup. The first step is to set the Fmax. The Fmax setting determines the highest frequency in the spectrum. It is important to realize that this will be the low pass filter applied to the waveform, and that the waveform sample rate will be 2.56 times this number as well.



With the analyzer in the **Spectra Analyze Mode**, set an Fmax of **800 Hz**. The waveform sample rate will be 2048 Hz. (800 x 2.56 = 2048 samples per second)

Next, consider the number of lines of spectral resolution and how this affects the waveform. For this example, set the number of lines to **800**. The 800 Hz Fmax will be divided equally into 800 lines (or bins) of resolution. Therefore, each bin of resolution will be 1 Hz wide. However, the selected number of lines of resolution also has an effect on the time waveform.

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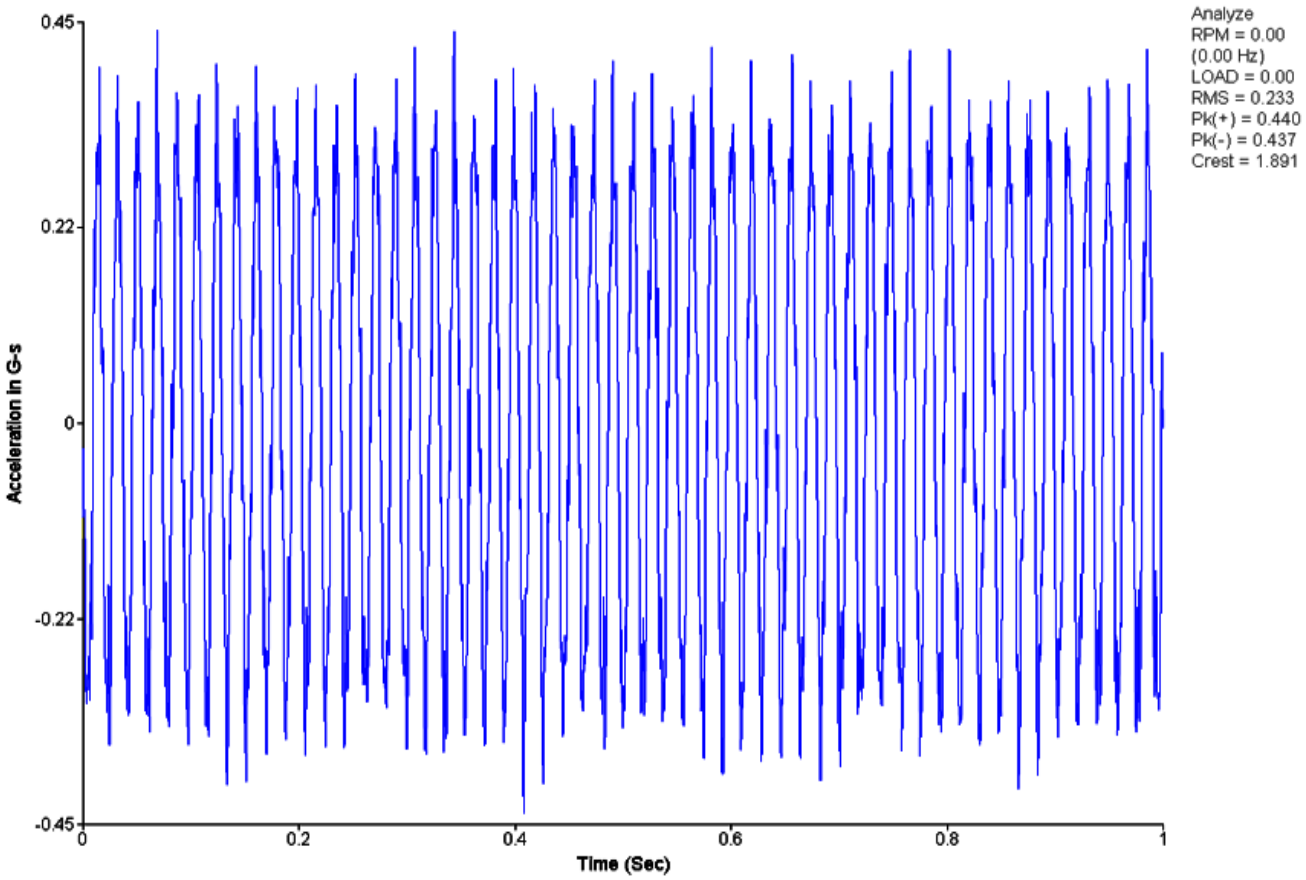
The length of the time waveform can be calculated as the number of lines divided by the Fmax. If frequency units of Hz are used for the Fmax, then the time will be in seconds. If frequency units of CPM are used, then the result will be in minutes. In this case, the resulting time waveform is exactly one second. (800 lines ÷ 800 Hz = 1)



Remember, the sample rate for this waveform was 2048 samples per second. Since this waveform is exactly one second long, then this waveform must be made up of exactly 2048 data points.

To increase the time length of the waveform, the analyst can elect to either decrease the Fmax or increase the number of lines of resolution used in the spectrum.

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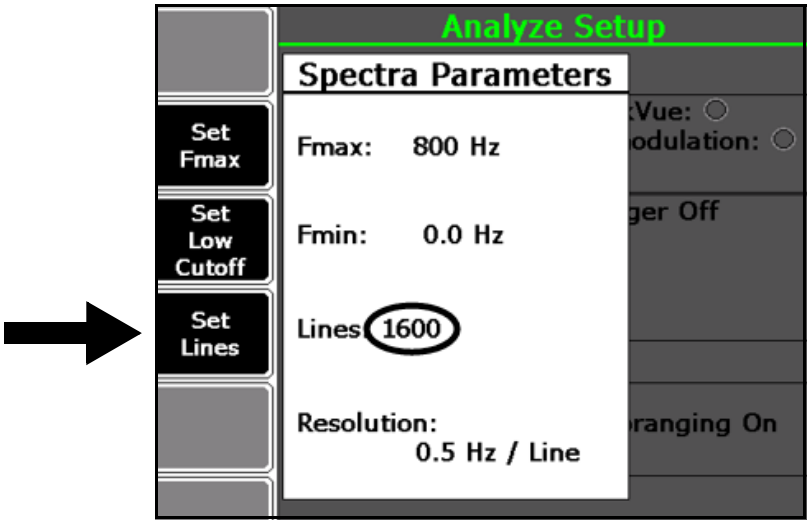
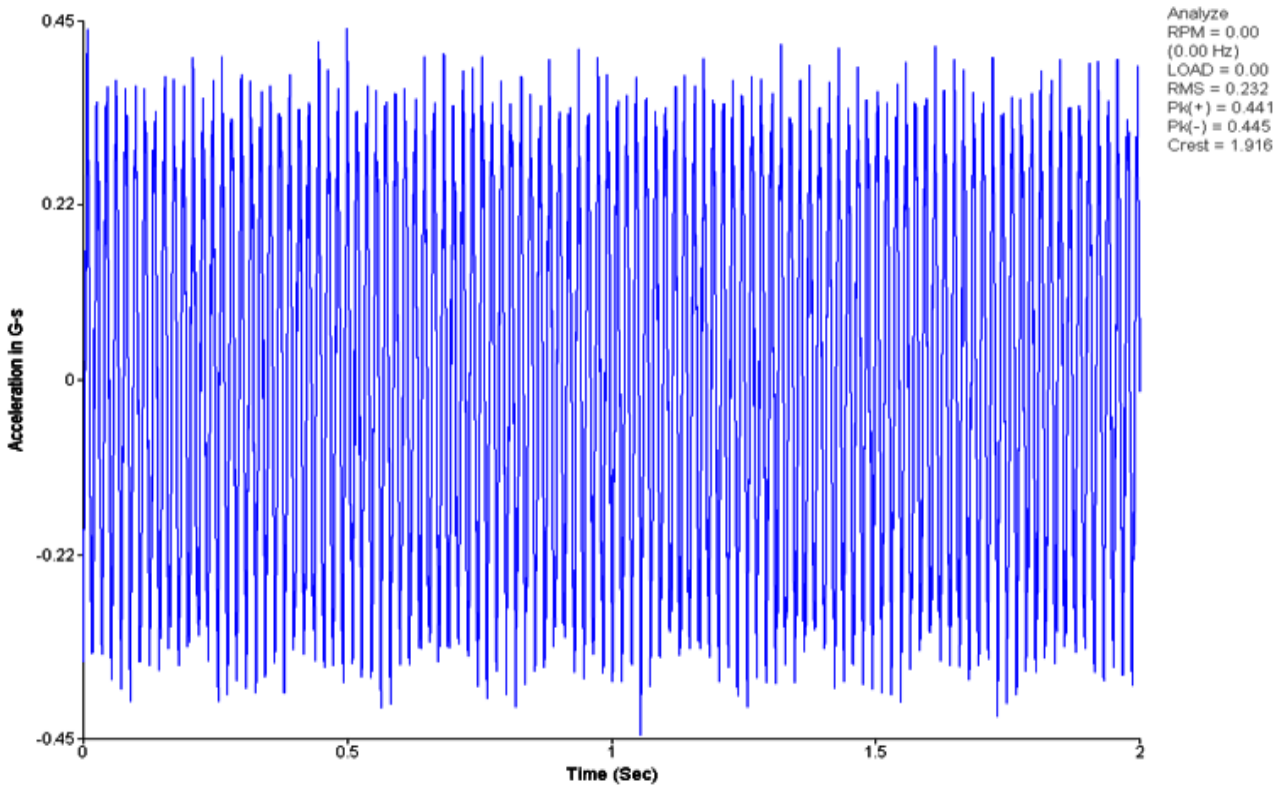
For this example, increase the lines to **1600**. The effect on the spectrum is a bin width that has been reduced from 1 Hz per line to .5 Hz per line.



The Fmax is unchanged for this waveform so the sample rate is still 2048 samples per second. However, the waveform is now twice as long as the previous waveform. The number of samples it contains is 4096. (2048 X 2 sec)



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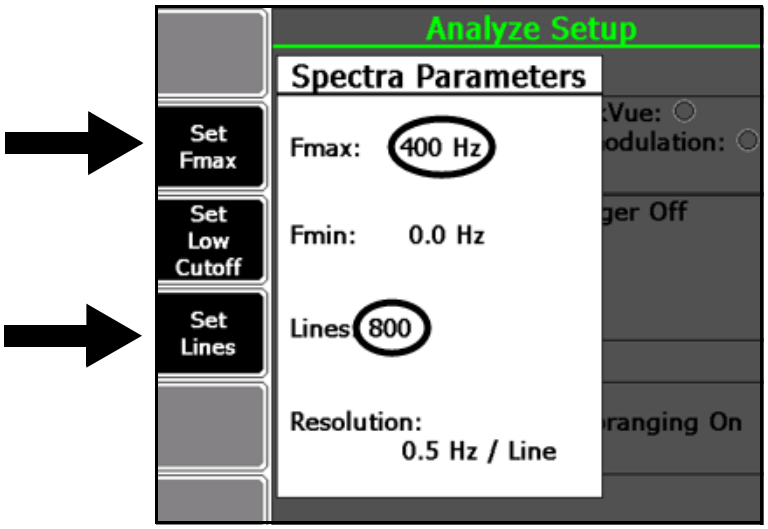
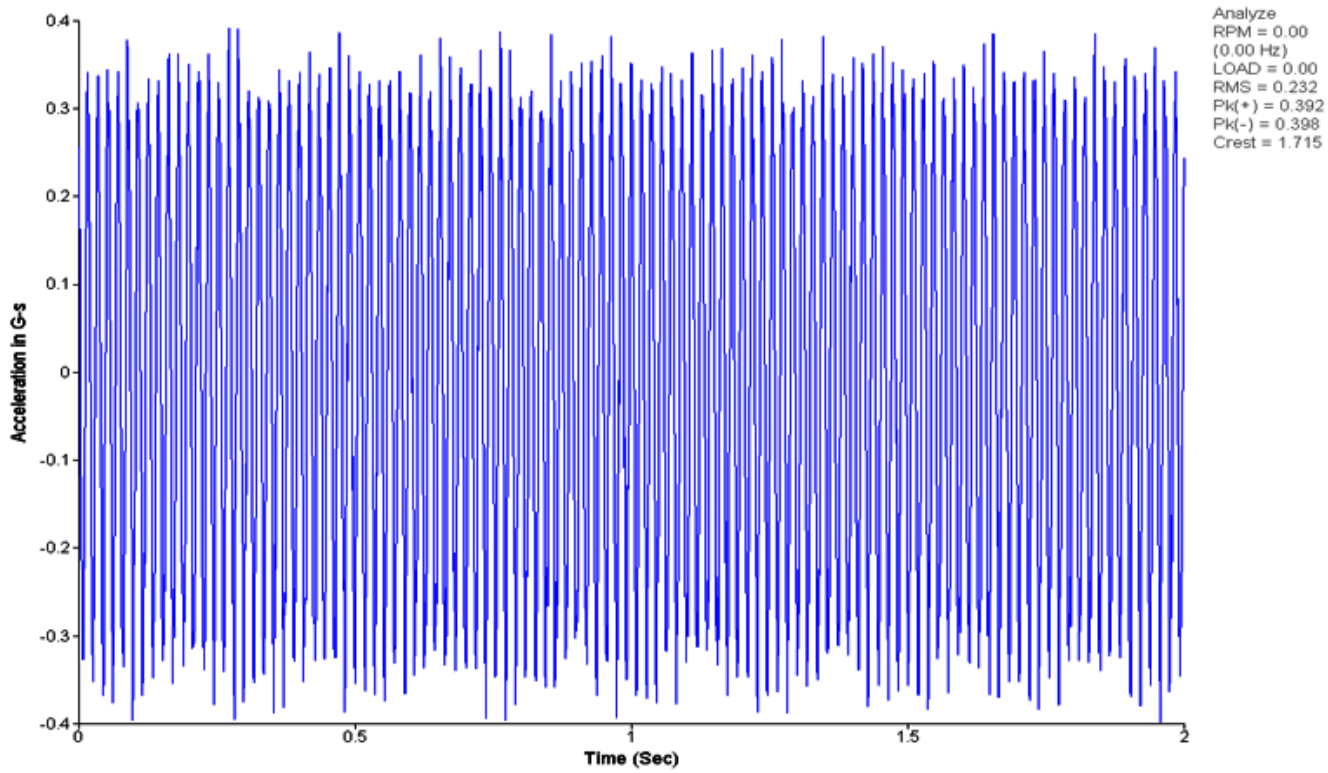
Next, decrease the Fmax to 400 Hz, and set the spectral resolution back to 800 lines. Based on the lines divided by Fmax equation, this should also result in a two second waveform. The spectral resolution will also remain at a bin width of .5 Hz.



Although this waveform is still two seconds in time, the sampling rate was half that of the previous waveform. Instead of sampling 2048 times per second, this data was only sampled 1024 times per second, leaving 2048 samples.



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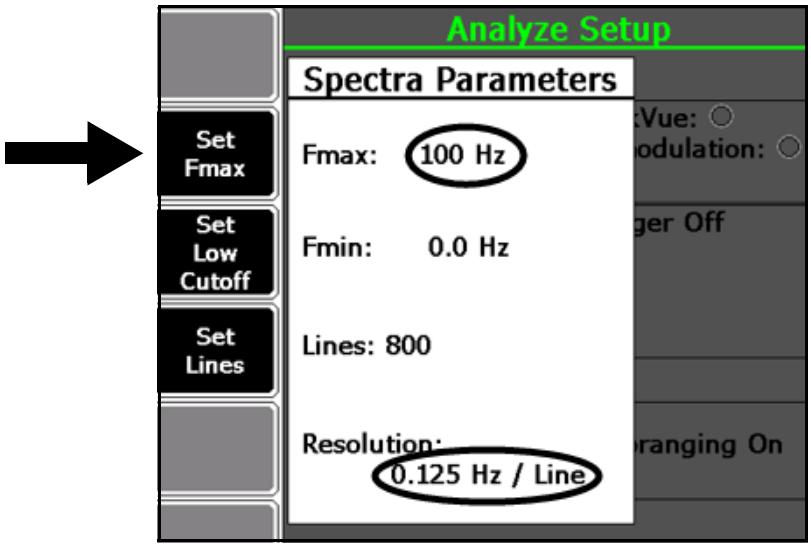
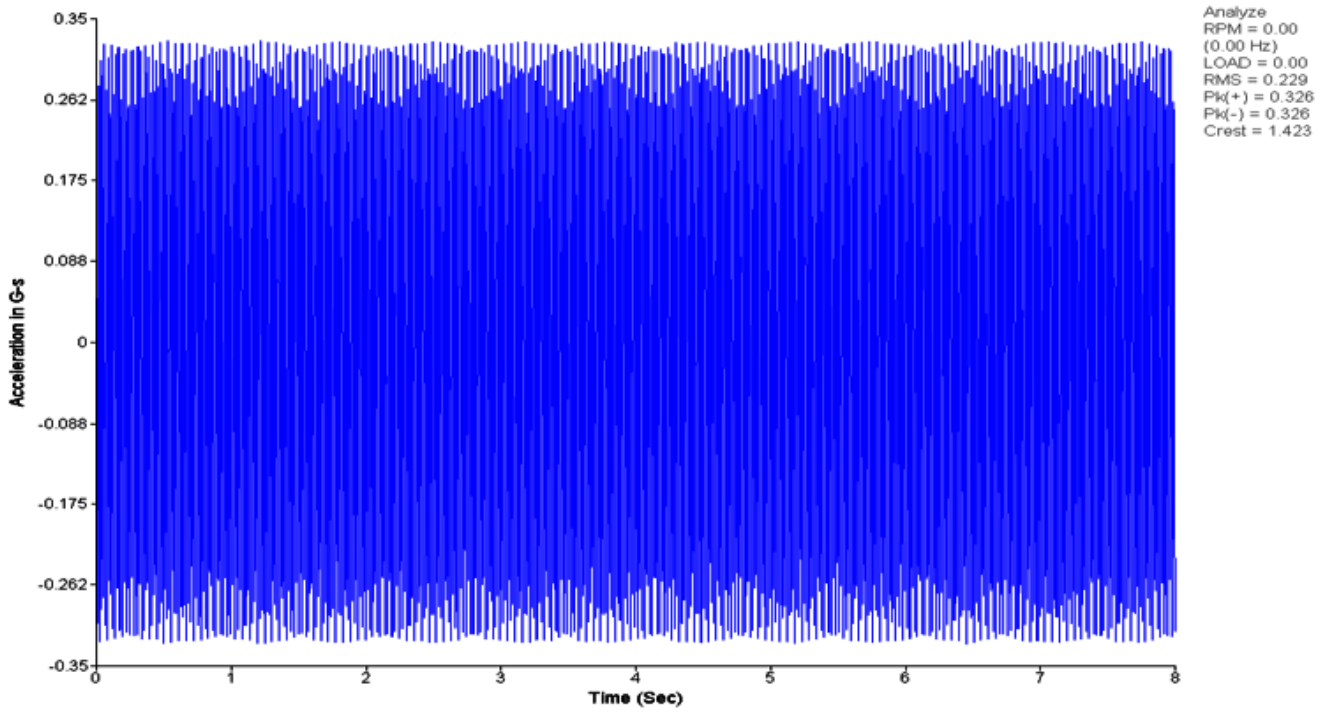
Now, bring the Fmax down to 100 Hz. Notice, the spectral resolution is now down to a bin width of only .125 Hz per line. This will result in a waveform that is eight seconds in time.



However, the total number of data points that it contains is still 2048. The decrease in Fmax from 400 Hz to 100 Hz has decreased the sample rate from 1024 samples/sec to only 256 samples/sec. This waveform has poorer resolution than those acquired using a higher Fmax, since lowering the Fmax lowers the sample rate.



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Time Waveform Parameters

Because of the relationship between the Fmax, sample rate and the lines of resolution, whenever a spectrum is acquired using 800 lines of resolution, the corresponding waveform will have 2048 data points. Another way to calculate the number of data points contained in the waveform is to multiply the lines of resolution by 2.56.

- 400 line spectrum requires 1024 waveform data points.

  (400 x 2.56 = 1024)

- An 800 line spectrum requires 2048 waveform data points.

  (800 x 2.56 = 2048)

- A 1600 line spectrum requires 4096 waveform data points.

  (1600 x 2.56 = 4096

- A 3200 line spectrum requires 8192 waveform data points.

  (3200 x 2.56 = 8192)

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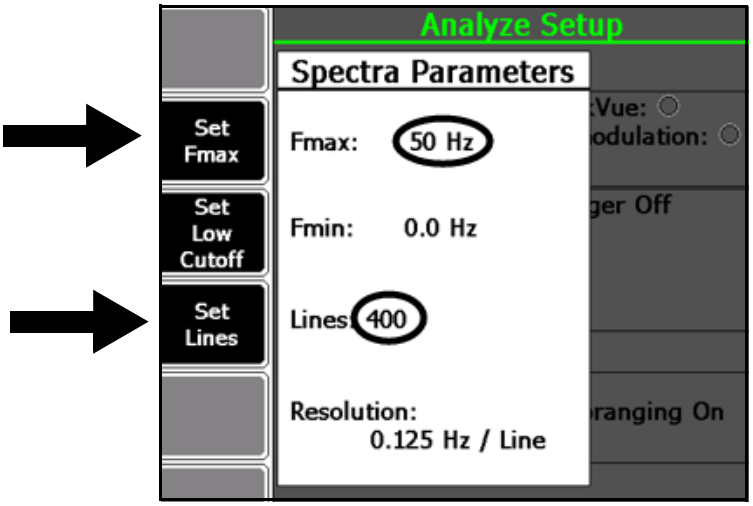
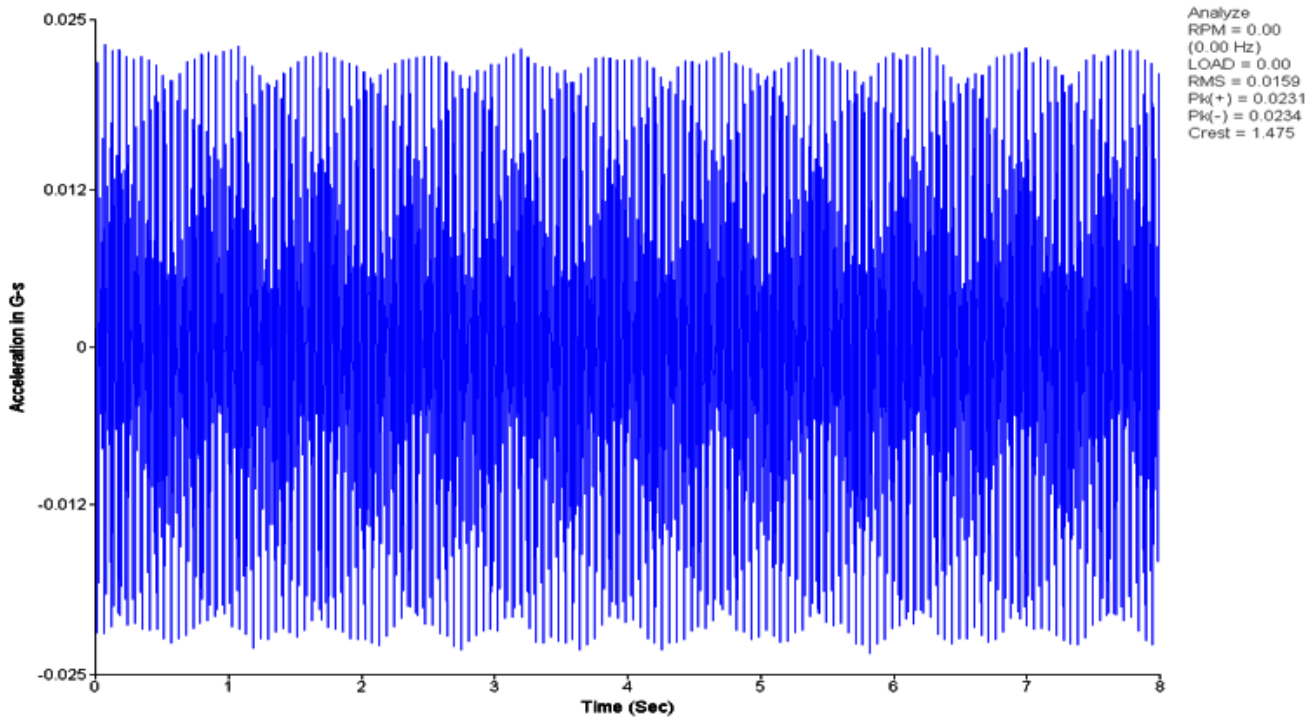
Finally, bring the Fmax down to 50 Hz, and set the number of lines to 400. This will still result in an eight second waveform, and a bin width of .125 Hz. Notice one very large difference.



The waveform amplitude in G's has decreased from about .8 G's Pk-Pk to about .05 G's Pk-Pk. The reason being that the Fmax of 50 Hz is now below the rotor kit's turning speed of around 53 Hz. The anti-aliasing filter has filtered out most of the vibration energy.



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Fast Fourier Transform

The conversion of time domain information to frequency domain information is the **Fast Fourier Transform (FFT)**. Often a frequency spectrum is referred to as an FFT. The FFT is the mathematical conversion from the time domain to the frequency domain. This math process makes several assumptions about the waveform data. One assumption that is effectively true is that the waveform was happening before the data was acquired, and it will continue in the same way after the data is has been acquired. (It is periodic.) This is true for industrial rotating equipment. Another assumption implied in the FFT process, is the sample waveform is composed of many sine waves at different frequencies.

This turns out to be true sometimes and not so true at other times, depending on the type of defect being analyzed.

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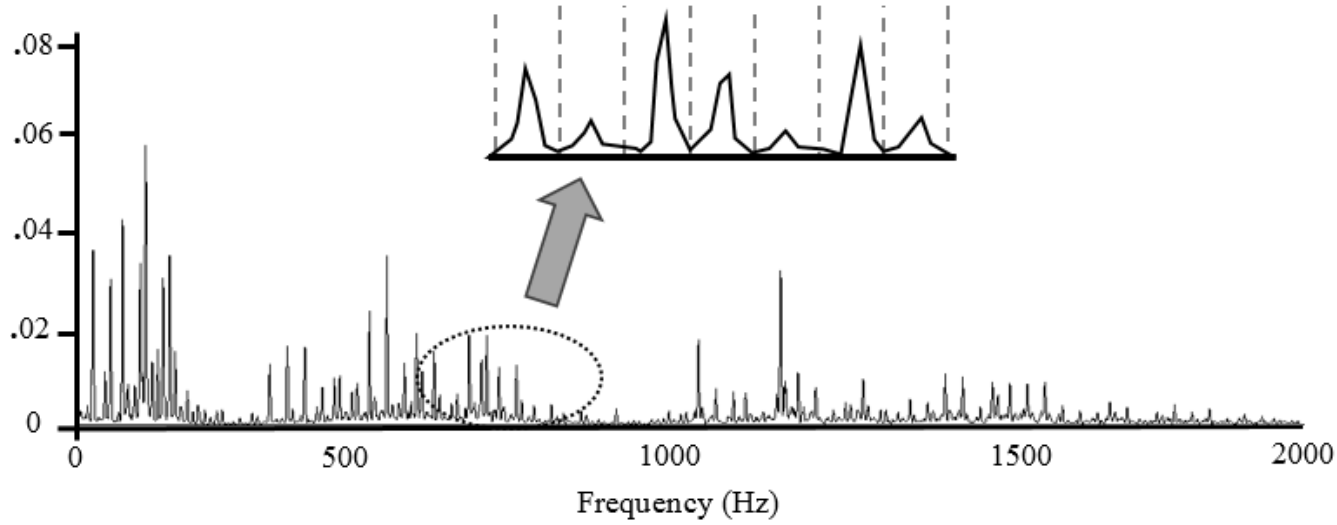
Spectral Resolution

Like the digital waveform, the digital spectrum also contains discrete areas at which to place the data. **Resolution** is the number of parts of the spectrum, usually called bins, or lines of resolution (LOR). The number of LOR selected is divided into the maximum analysis frequency (Fmax) to arrive at the bin width (BW).

BW = Fmax/ LOR



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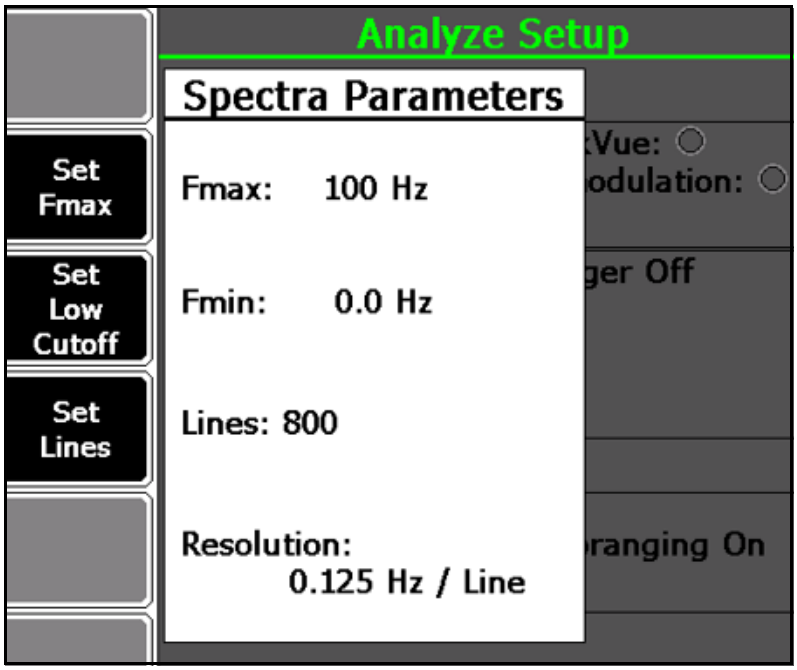

Notice that while setting up spectra parameters in the analyzer, the Bin Width is calculated for you. Remember, bin width is the Fmax divided by the number of lines. Also, remember that the number of lines divided by the Fmax equals the length of time in the time waveform.

It is important to realize that the peaks in the spectrum are actually the center frequencies of these bins of energy. Each bin contains an infinite number of frequencies, and all the energy in the bin is summed and represented by a single amplitude at the center frequency of the bin.

The digital analyzer allows the analyst to set a spectrum maximum frequency (Fmax) and a number of lines of resolution (100,200,400,800, etc). This implies that the spectrum can only display data at a frequency located at each bin, and any data that is close together in frequency will be placed together in the bin, resulting in the inability to distinguish closely spaced frequency defects. In short, **the spectral accuracy depends entirely on the spectral resolution**.

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Often, analysts use order based Analysis Parameter sets in their equipment databases, and the same AP set gets used on many various pieces of equipment. Consider the effect of RPM on the BW with a typical 70 x RPM Speed Rolling Element Bearing Set.

RPM	x 70 = Fmax	/LOR	= BW(Hz)
20 Hz (1200 CPM)	1400	800	1.75
30 Hz (1800 CPM)	2100	800	2.65
60 Hz (3600 CPM)	4200	800	5.3

Notice, that use of the same AP set will result in the higher speed equipment getting data with much less spectral resolution than the lower speed equipment. Therefore, it is recommended that different AP sets be created and used for machines running at different speeds.

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Effective Resolution

Since most spectral data that gets acquired uses the Hanning window, the effective resolution, or effective bin width, is not as simple at dividing the Fmax by the lines of resolution. An additional factor that needs to be considered in this equation is called **Window Noise Factor**. In order to more accurately calculate the spectral accuracy, the simple bin width needs to be multiplied by the **window noise factor**. **The window noise factor for Hanning window is 1.5**. The calculation is below:

Effective Resolution = BW x 1.5

Use the Bin Width from the previous example to calculate the Effective Resolution:

Effective Resolution = .125 Hz x 1.5

Effective Resolution = .1875 Hz

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How Much Resolution is Enough?

In order to accurately identify both the frequencies and the amplitudes of two closely spaced peaks in the spectrum, there needs to be at least one empty (or mostly empty) bin of resolution between them. The bin width needs to be equal to or less than half of the difference frequency of any closely spaced peaks. The **window noise factor** needs to be figured into this as well. In order to calculate the minimum number of lines needed, use the following equation:

$$\textbf{Minimum \# FFT lines} \geq \frac{\textbf{2 * window noise factor * Fmax}}{\textbf{difference frequency}}$$

Consider the following situation: The analyst is measuring vibration on a two pole motor running 3580 RPM. The route spectrum taken with an Fmax of 4250 Hz uses 1600 lines of resolution. The analyst sees what looks like a peak of vibration at 2x RPM in the spectrum. As is common with two-pole motors, this peak may be vibration at 2x line frequency or it may be excessive vibration at 2x RPM due to misalignment. Does the analyst have enough effective resolution in the spectrum in order identify the true source of the vibration?

First, calculate the difference frequency.

2x RPM = 3580 x 2 = 7160 CPM or 119.33 Hz

2xLF = 60 Hz x 2 = 120 Hz

Difference Frequency = 120 - 119.33 = 0.67 Hz

Effective Resolution = 4250 x 1.5 ÷ 1600 = 3.98 Hz

The analyst will not be able to determine the frequency or amplitude of each peak.

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The two peaks are separated by only 0.67 Hz, and the effective resolution is 3.98 Hz. Now, calculate the minimum number of lines needed:

$$\textbf{Minimum \# FFT lines} \geq \frac{\textbf{2 * window noise factor * Fmax}}{\textbf{difference frequency}}$$

$$\textbf{Minimum \# FFT lines} \geq \frac{\textbf{2 * 1.5 * 4250}}{\textbf{0.67}}$$

$$\textbf{Minimum \# FFT lines} \geq \textbf{19,030}$$

Since the maximum number of lines available on a CSI 2130 is 12,800, the analyst needs to lower the Fmax. Calculate the minimum number of lines needed using an Fmax of 400 Hz.

$$\textbf{Minimum \# FFT lines} \geq \frac{\textbf{2 * 1.5 * 400}}{\textbf{0.67}}$$

$$\textbf{Minimum \# FFT lines} \geq \textbf{1,791}$$

In this case, the analyst would need to acquire a spectrum using an Fmax of 400 Hz and 3200 lines of resolution in order to identify just how much vibration was being caused by electrical sources and how much was being caused by mechanical sources.

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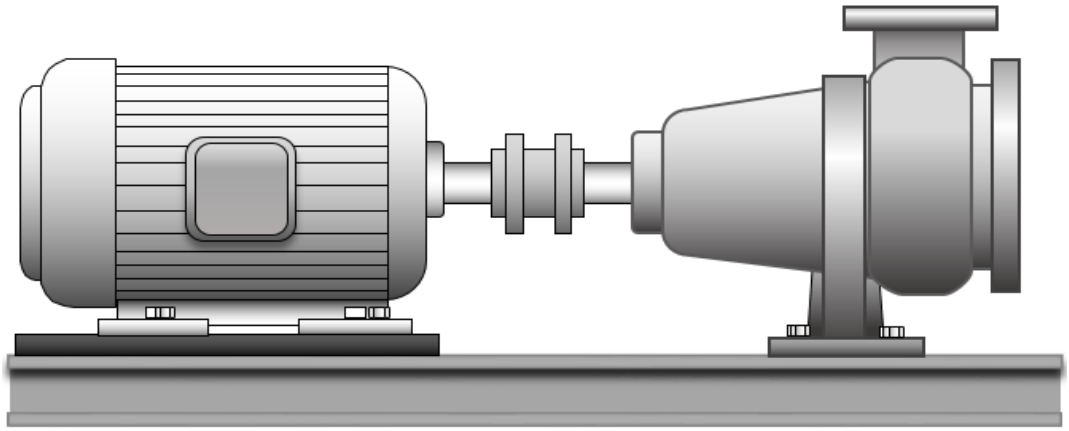
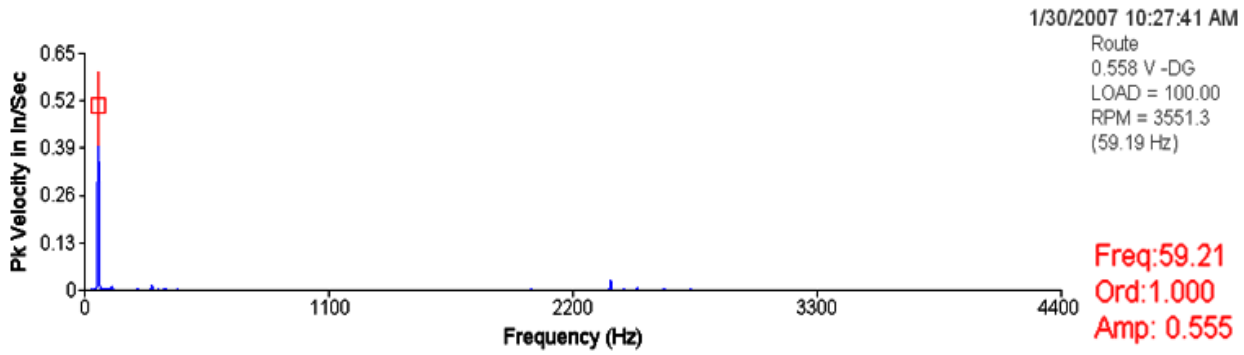
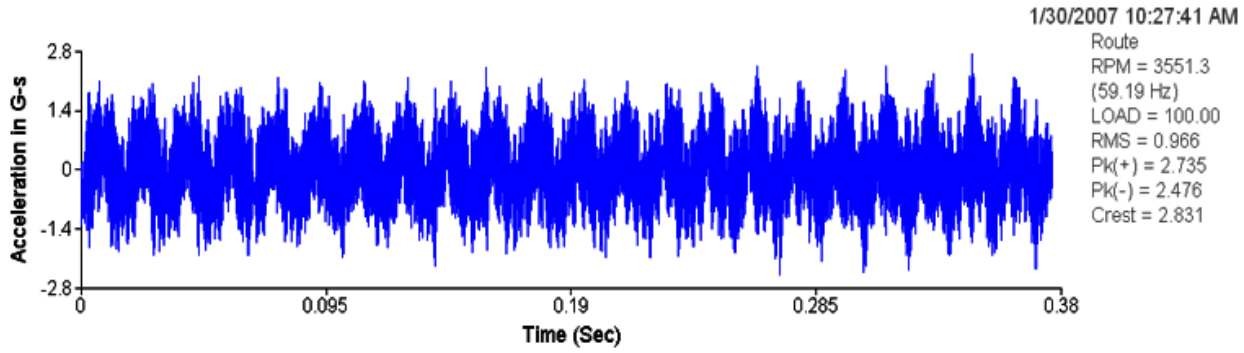
Spectral Resolution Case History



Boiler Feed Pump #1 - RPM = 3550
In-line coupled overhung pump





Normal route data calls for a spectral Fmax of 4,250 Hz. 1600 lines of resolution are used netting a bin width of 2.65 Hz. The time waveform is .37 sec and contains 4096 data points. The data presented here is all from the motor outboard horizontal measurement point.

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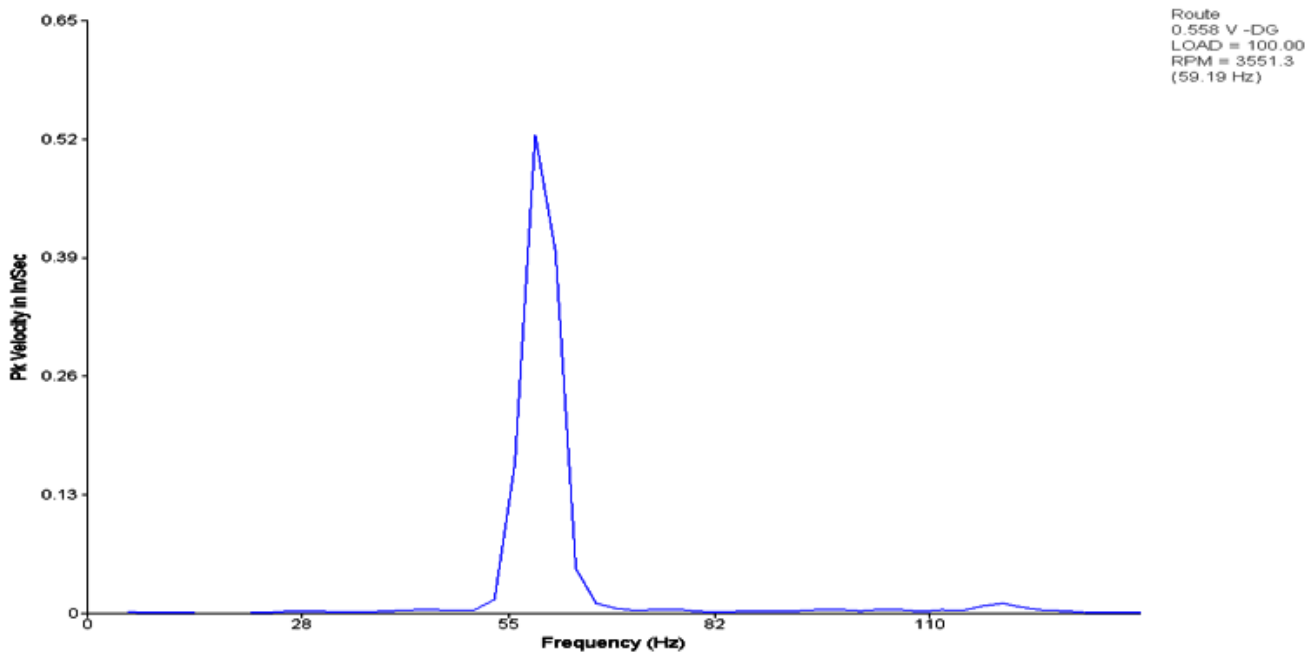
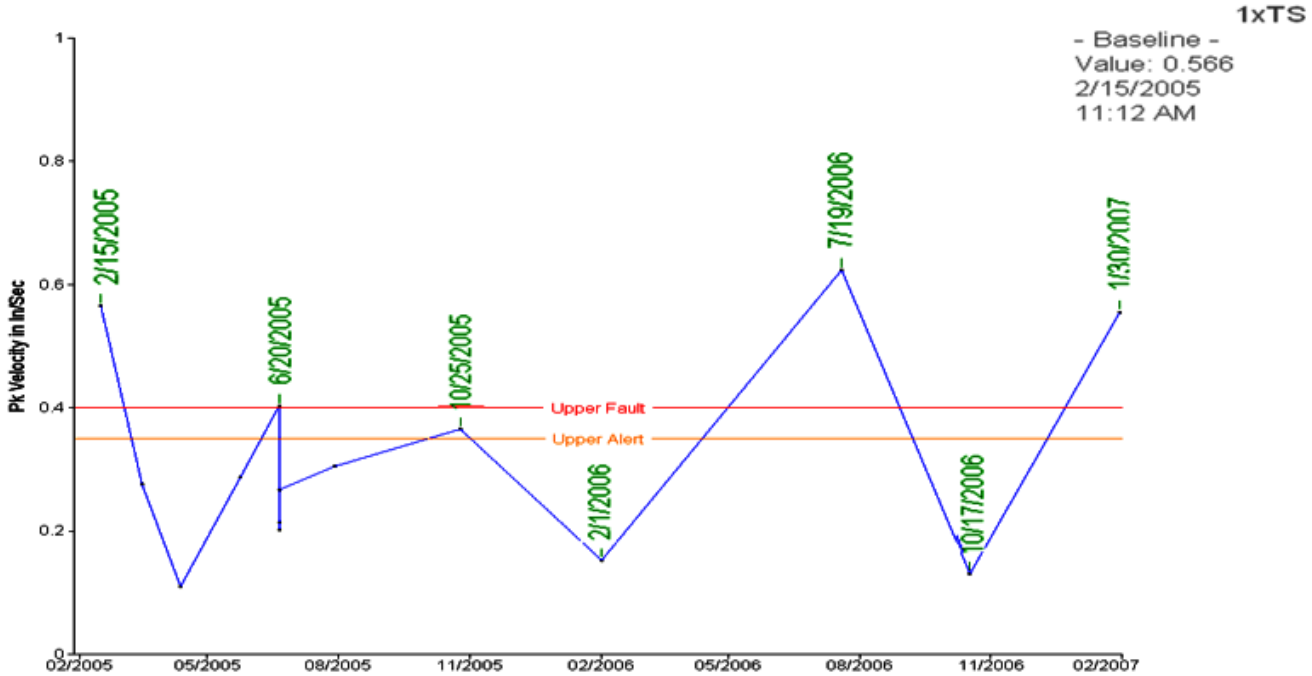
Using the horizontal axis zoom control, a zoom of the peak at 59 Hz appears normal.



The pump had experienced multiple failures in the previous two years. A trend of the vibration at 1X turning speed is very erratic. The amplitude ranges from .1 in/sec all the way up to .6 in/sec.



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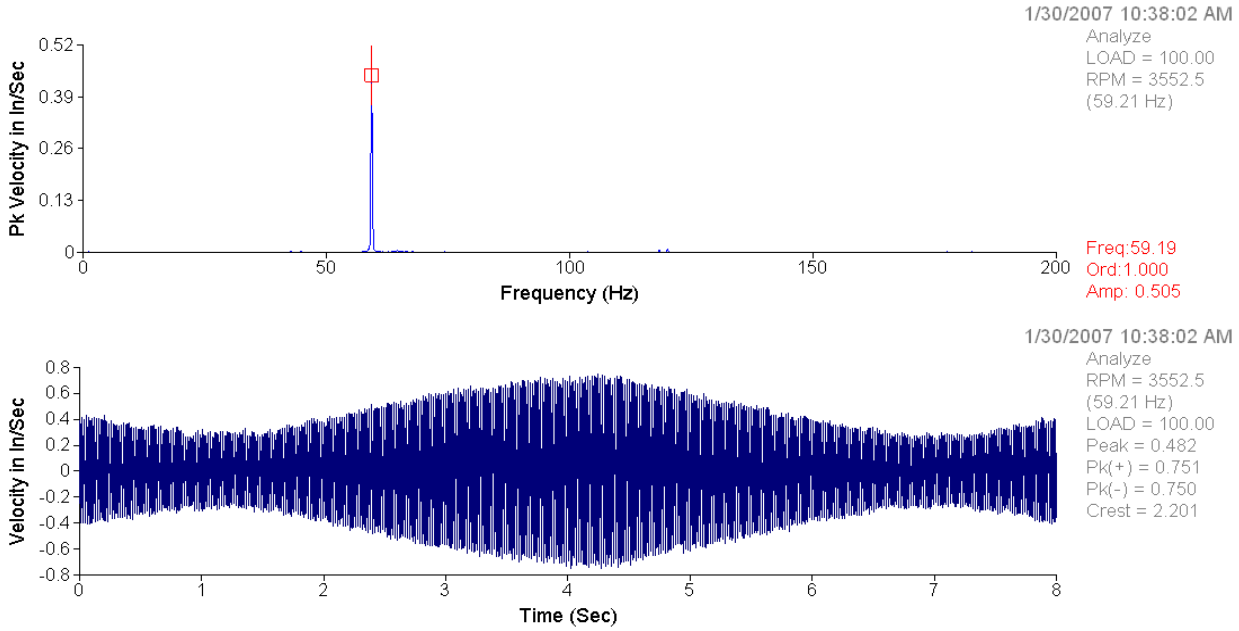
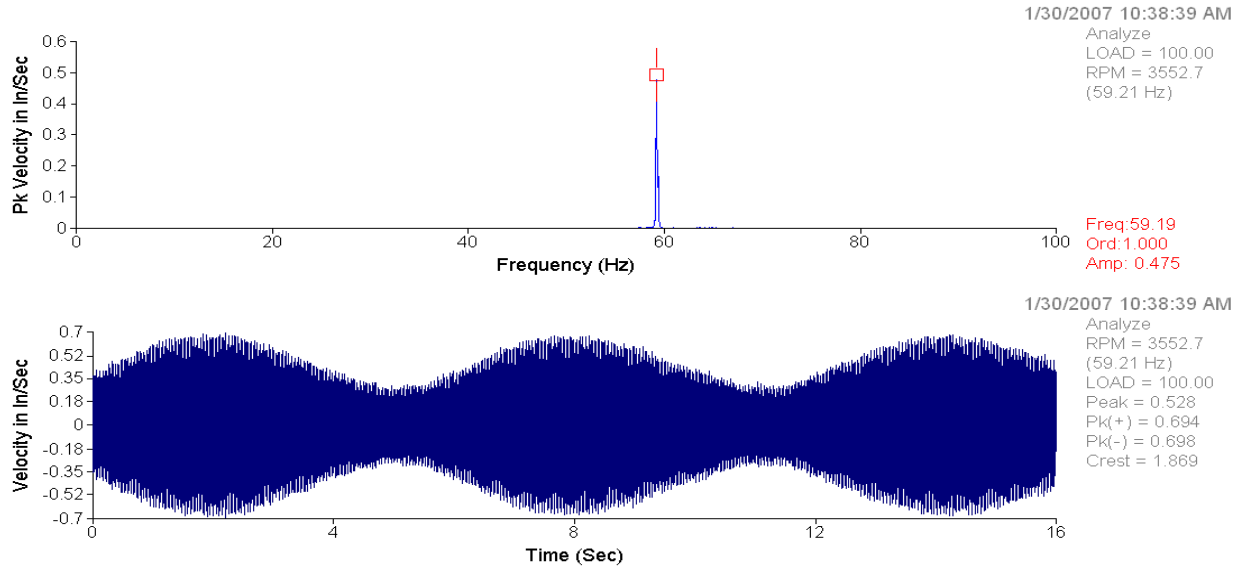


Additional data was acquired. This time, the Fmax was lowered to 200 Hz using 1600 lines of resolution. The bin width was reduced to .125 Hz, and the time waveform was increased to eight sec. It is now apparent that there is a change in amplitude (modulation) in the waveform.



The next question that needed to be answered was whether or not the amplitude change in the waveform occurred periodically. In order to acquire a longer waveform, the Fmax was lowered again, this time to 100 Hz. Here, the change in amplitude does appear to be periodic.

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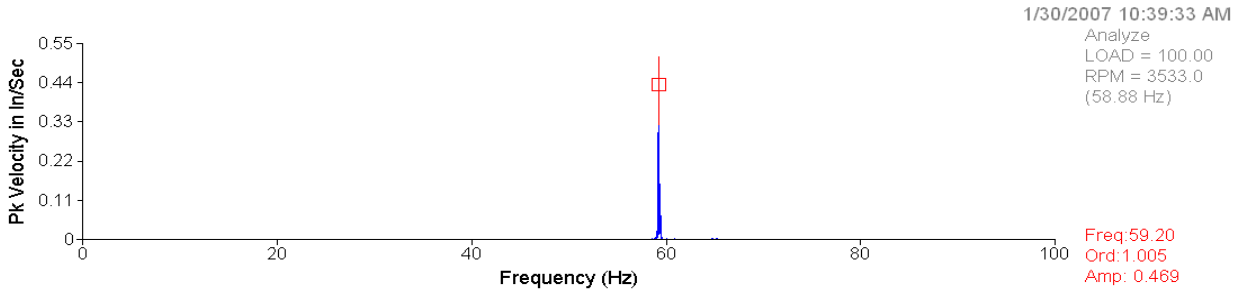
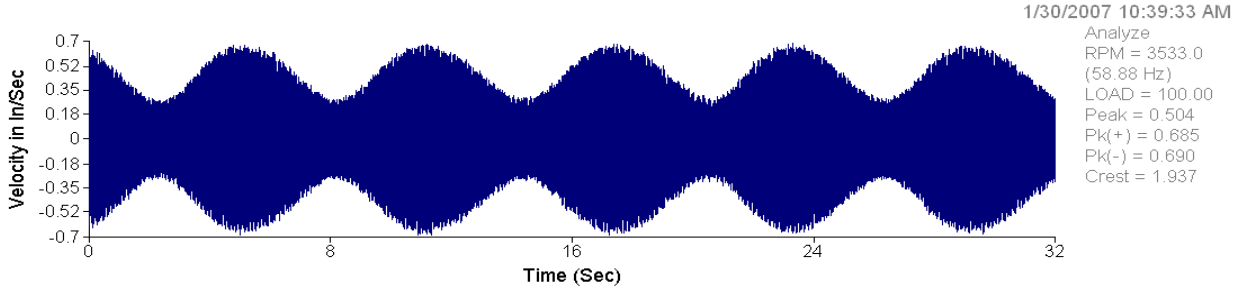
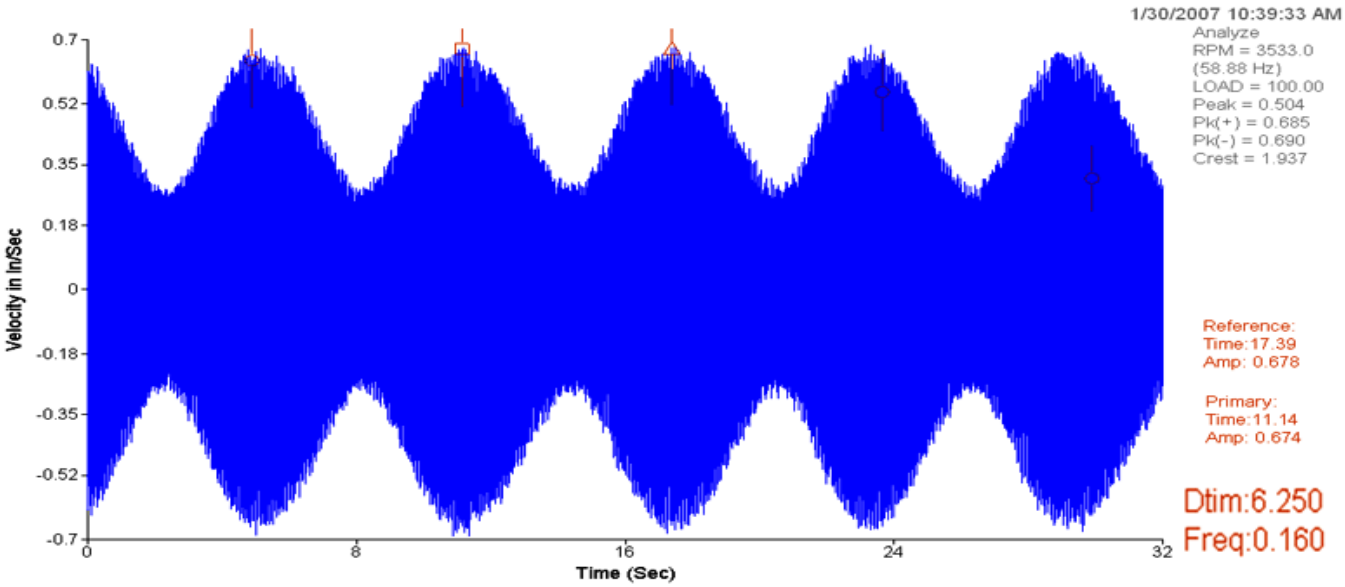




The analyst felt that even more time was needed in the waveform in order to assess the periodicity of the change in amplitude. Since most of the vibration amplitude is at 59.2 Hz, lowering the Fmax to 50 Hz was not an option. The number of lines was increased from 1600 to 3200 lines, resulting in a spectral bin width of .031 Hz and a time waveform of 32 seconds.



Using the difference cursor, we can see that the maximum amplitude in the waveform repeats about every 6.25 seconds. This would translate to a frequency of .16 Hz. (1/ 6.25 = .16)

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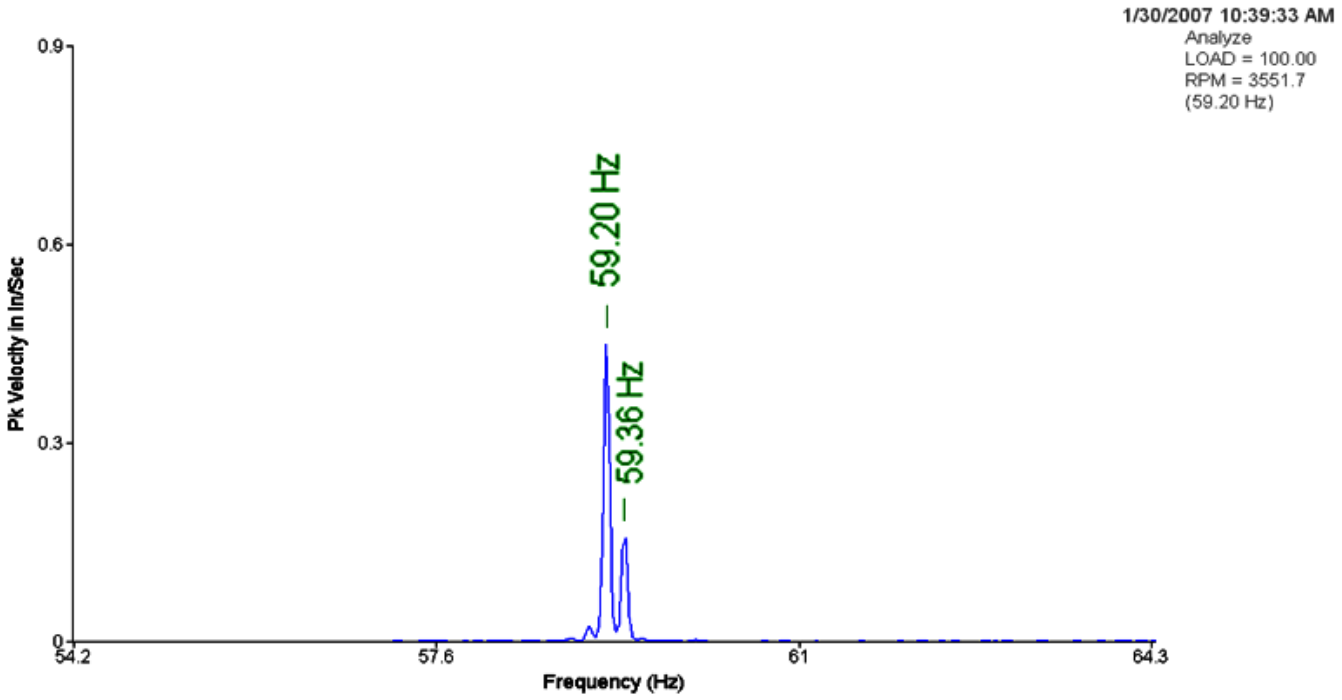

A zoom of the high resolution spectrum now reveals two distinct peaks. One is at 59.2 Hz (the turning speed of this shaft) and the other at 59.36 Hz. The difference in frequency between these two peaks is .16 Hz, which corresponds to the 6.25 second period of the periodic change in amplitude of the waveform.

The second peak in the spectrum is actually coming from another boiler feed pump that is mounted to the same base as number one. Since both motors turn at nearly the same speed, (the speed difference is 9.6 rpm) and are mounted to the same base, their vibrations go in and out of phase causing a vibration beat. The period of the beat (6.25 sec) corresponds to the difference in frequency between the two vibrations. (1/ 6.25 = .16)

Remember that the original route data used a spectrum with a bin width of 2.65 Hz, and a waveform that was only a fraction of a second in time. The analyst would not be able to identify the beat using the route data alone. The ability to manually manipulate the properties of both the waveform and the spectrum is something that every vibration analyst needs to develop.

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Waveform Shaft Rotations

For many diagnostic situations, it is advantageous to capture 10 to 15 revolutions of the shaft, or more, in the time waveform. In order to calculate the amount of shaft revolutions contained in the time waveform, one might find it useful to first calculate the time period of one shaft revolution in seconds. This is done by first dividing the shaft RPM by 60 in order to get the shaft RPS (rotations per second) Second, calculate the inverse of the shaft RPS.

As an example use 1800 rpm:

1800 rpm = 30.0 Hz

1 divided by 30.0 = .033 sec/ 1 rev

One revolution of the shaft takes .033 seconds.

.033 sec x 15 = .495 sec

A waveform of .495 seconds would be needed to capture 15 shaft revolutions of the shaft at 1800 RPM.

The typical PDM based AP set for machines with antifriction bearings acquires an Fmax of about 70 orders, using 1600 lines of resolution. Will this setup provide a time waveform that captures 15 shaft revolutions if the shaft RPM is 1800?

Fmax = 70 (orders) x 30Hz (which is 1800 CPM)

Fmax = 2100 Hz

1600 Lines divided by 2100 Hz = .76 second waveform

This setup would indeed be appropriate for use with an 1800 RPM anti-friction bearing application.

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An easier way to calculate the shaft revolutions in the time waveform is to use the previously mentioned calculation for time waveform length, which is number of lines of resolution divided by the spectral Fmax.

waveform length=# lines ÷Fmax

If the value used for Fmax is in CPM, the answer for time waveform length will be in minutes. If the value used for Fmax is in Hz, then the answer will be in seconds. If the value used for Fmax is in Orders, the answer will be in shaft revolutions. From the previous example, the time waveform in minutes would be:

1600 ÷ 126,000 = 0.0126 minutes

In seconds, the time would be:

1600 ÷ 2,100 = .76 seconds

For shaft revolutions, the calculation would be:

1600 ÷ 70 = 22.8 revolutions

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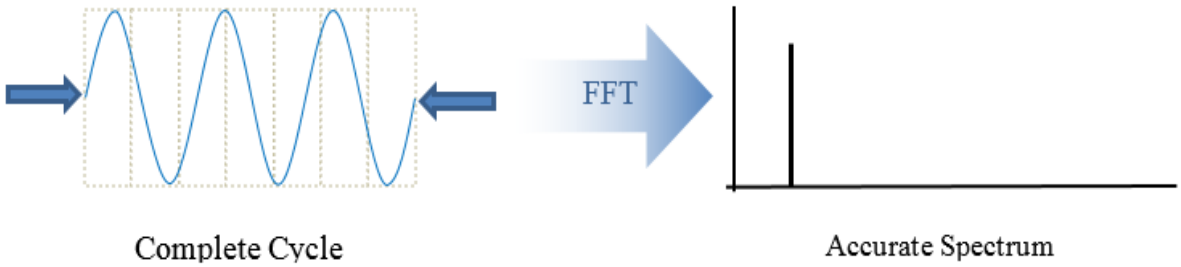
Leakage, Hanning Window and Data Overlap Averaging

Leakage

Analyzing the spectrum to diagnose machinery faults requires accurate frequency and amplitude calculations. It should be noted that the spectrum is a calculation, not an actual measurement. The waveform represents the actual measurement.

The peaks in the spectrum are calculated from the periods of cycles in the waveform using Fast Fourier Transform (FFT). The FFT process assumes that the waveform is a finite set of data. The calculations are based on a time domain consisting of one continuous periodic signal. Therefore, FFT interprets the two end points as though they are connected together. The FFT process will treat the calculation as though each period in the waveform sample has a start and end point resulting in discrete individual frequencies.

Below is an example of a periodic signal that has an exact integer of one period. An FFT calculation would therefore produce a spectrum with a single peak representing this frequency.



In the example above, the period of this cycle would be calculated as a complete single frequency because the signal starts and ends in a position that allows a continuous single frequency to be measured. The same would occur if there were a large number of periods as long as the end points allowed for continuous cycles during the calculation.

When the FFT process is applied to a waveform that has a non-integer multiple of periods, (incomplete cycles), the calculation will result with frequencies spread into a wider signal. This will cause the calculated energy to spread into adjacent bins or lines of resolution. This is known as spectral **leakage**.

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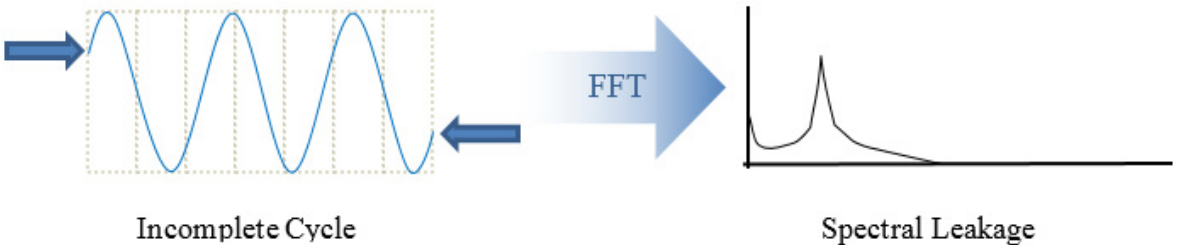
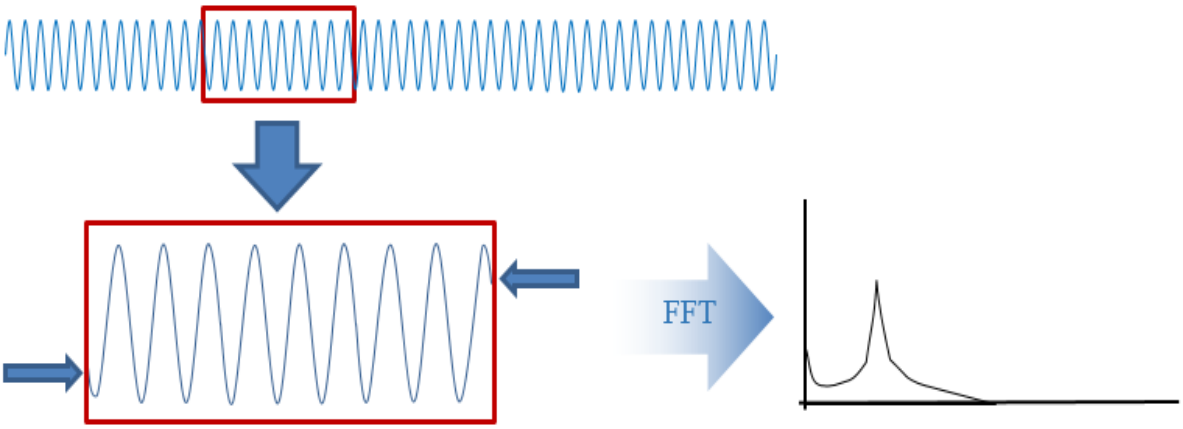
Below is an example of a periodic signal that does not contain an exact integer of a period. Applying FFT to this signal would produce a spectrum with a smeared peak, instead of a single frequency.



When monitoring machinery vibration, acquiring a waveform with an integer number of periods is practically impossible. The data acquisition is a short time block of the continuous vibration in the machine. The length of this time block is predetermined by settings in the analysis parameter set. Controlling where the first and last sample will be measured in the waveform is not achievable. Consequently, the waveform will contain incomplete cycles, resulting in a spectrum with excessive noise floor.

This image below shows one time block measured from the continuous signal of the machine vibration.



Each waveform that is captured during normal route data collection will contain many cycles. This can increase the amount of leakage in the spectrum.

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Windowing

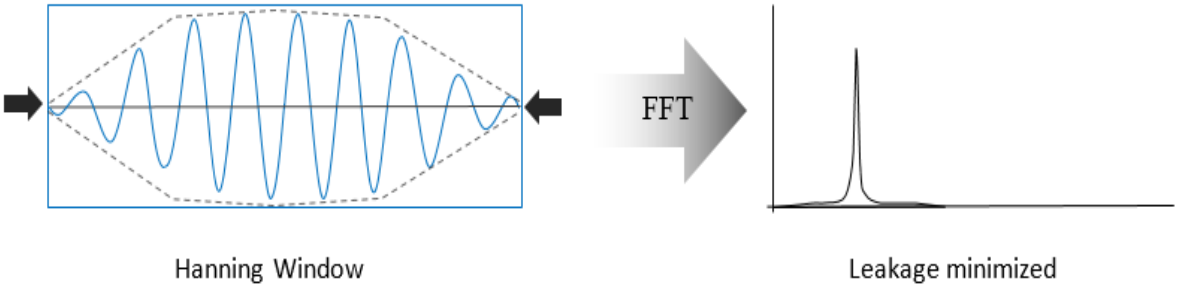
To minimize spectral leakage, a technique called **windowing** is applied to the waveform. The windowing function normally improves the results of the frequency domain.

There are several types of windows that may be applied to the waveform. The various window types have different effects on the data and are used for various tests or measurements.

However, the Hanning window works best for bringing the end points of the waveform closest to zero, and is therefore applied in most vibration analysis applications.

Hanning Window

Using a cosine taper, the Hanning window reduces the amplitude of the end points of the waveform, tapering them gradually toward zero for the purpose of creating a continuous waveform without sharp transitions.



Other Window Types include **Uniform**, **Flat Top**, **Kaiser Bessel**, **Hamming**, **Force** and **Exponential**.

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Data Overlap Averaging

To acquire accurate frequency and amplitude data, the data acquisition should include enough of the vibration signal from the machine to get the best representation of the actual movement. Many times for normal route data collection, the length of the time waveform is less than one second. In most cases, a single waveform does not produce an accurate representation of the machine's vibration signature. A longer time record would render better results.

To achieve a longer time record, several waveforms are acquired to generate several spectra. The spectra are averaged to produce the final spectrum for analysis.

Data averaging is applied to the spectrum, not the waveform. Using more averages yields better data.

Since the Hanning window is applied to each of the waveform samples, the end points have been forced to zero amplitude. During the averaging process, the waveforms are overlapped to reduce the effect of any amplitude inaccuracies resulting from the Hanning window function.

The recommended overlap is 67 percent. After the acquisition of the first waveform, the analyzer stores 67 percent of the data in a time buffer. Each new average uses the last 67 percent of the data points from previous average. Therefore, each waveform average after the first consists of only 33percent new data, and is completed in approximately one third the time of the first average.

Thus, overlap averaging benefits the vibration analyst in two ways:

1. Data acquisition time is significantly reduced.
2. Amplitude inaccuracy caused by use of the Hanning window is minimized.

An example of overlap averaging using the Hanning window and five averages is shown on the next page.

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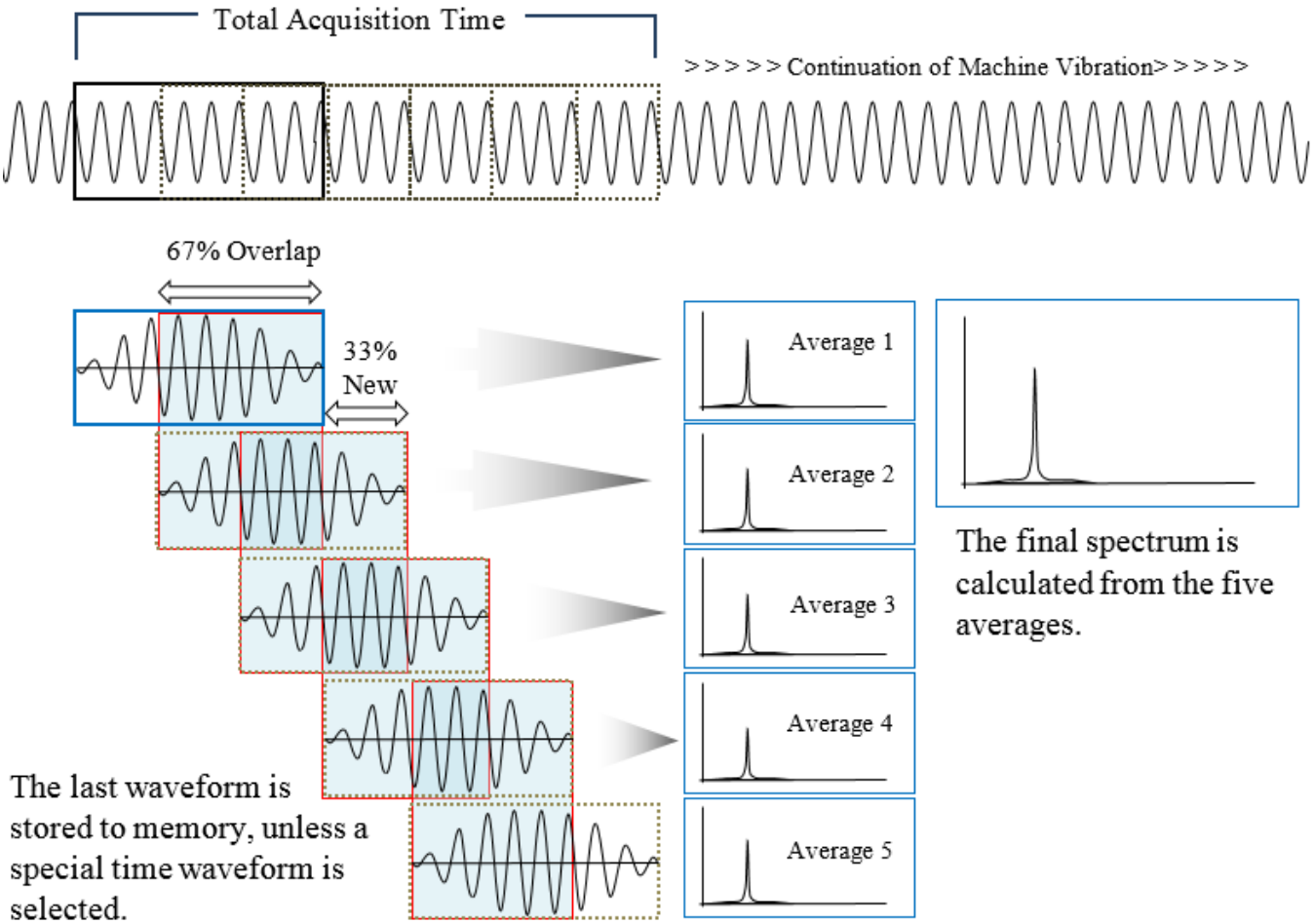

Time Required for Data Acquisition Using Overlap Averaging

The actual data collection total sample time is calculated by the formula:

$$T = (N - 1)(W)(1 - F) + W$$

Where:

T = data collection time

N = number of averages desired

W = time for one waveform average (LOR / Fmax)

F = Overlap Factor (percentage of overlap set in the analyzer)

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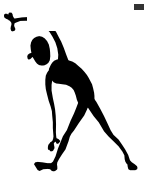

Data Acquisition Exercise

Calculate total data collection time necessary for a spectrum using the following parameters:

- Fmax = 1000 Hz
- LOR = 1600
- Window type = Hanning
- Number of averages = 6

The calculation should go like this:

$$\boldsymbol{T = (N-1)(W)(1-F)+W}$$

$$\boldsymbol{T = (5)(1.6)(.33)+1.6}$$

$$\boldsymbol{T = 4.24}\ \textbf{sec}$$

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Integration and Differentiation

Because of their low cost, high degree of reliability and ease of use, the accelerometer is the type of vibration probe that is most often used for normal periodic vibration data collection. Since accelerometers measure force in units of G's, the resulting waveform and spectrum are also in G's.

However, through the availability of hardware integration and differentiation, the user has the choice of viewing the stored spectral data in amplitude units of acceleration, velocity or displacement.

D (Displacement) = distance traveled by vibrating object

V (Velocity) = amount of Displacement/change in Time

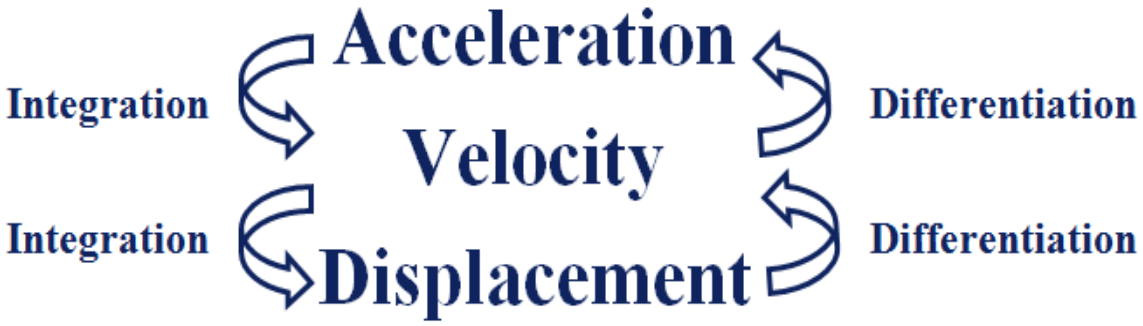
A (Acceleration) = amount of Velocity/change in Time

These terms are often represented with the following equations:

Velocity = D/ T (i.e. inches/second)

Acceleration = D/ T/ T = (i.e. inches/second/second)

Therefore, once any one of these amplitude units has been measured, integration and differentiation allows any of the other amplitude units to be calculated, provided the analyzer or software used is capable of this conversion process.



Converting acceleration values to velocity values is referred to as **integration**. The same term is used when converting velocity into displacement. The conversion from acceleration to displacement is referred to as **double integration**.

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Starting with displacement values and converting to velocity is called differentiation. Going from velocity into acceleration is also called **differentiation**. Starting with displacement and converting to acceleration is called **double differentiation**.

In most cases, the analysis of the spectrum is best performed using velocity amplitude units of inches per second. The reason is when velocity units are used, both the low frequency and high frequency amplitudes can be compared to each other. This comparison is not so easily accomplished when viewing the spectrum in either acceleration or displacement. When viewing the data in acceleration, the low frequency peaks are attenuated. When viewing the data in displacement, the high frequency peaks are attenuated. In order to view a velocity spectrum after an accelerometer was used to acquire an acceleration waveform requires two things:

1. **Integration must be performed to convert the acceleration values into velocity values.**
2. **FFT must be performed on the waveform to calculate the frequency spectrum.**

The order in which these two operations are performed is defined when the user chooses either **analog** or **digital** integration.

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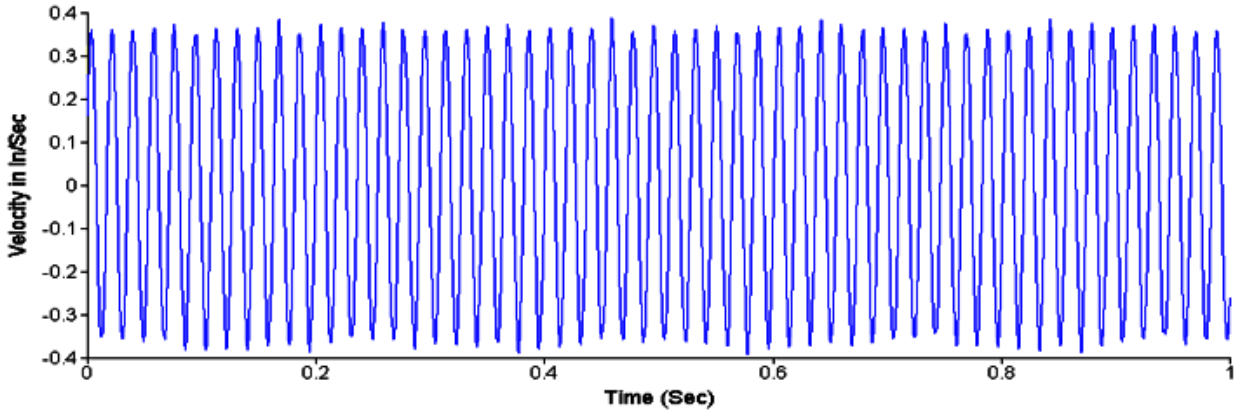
Analog Integration

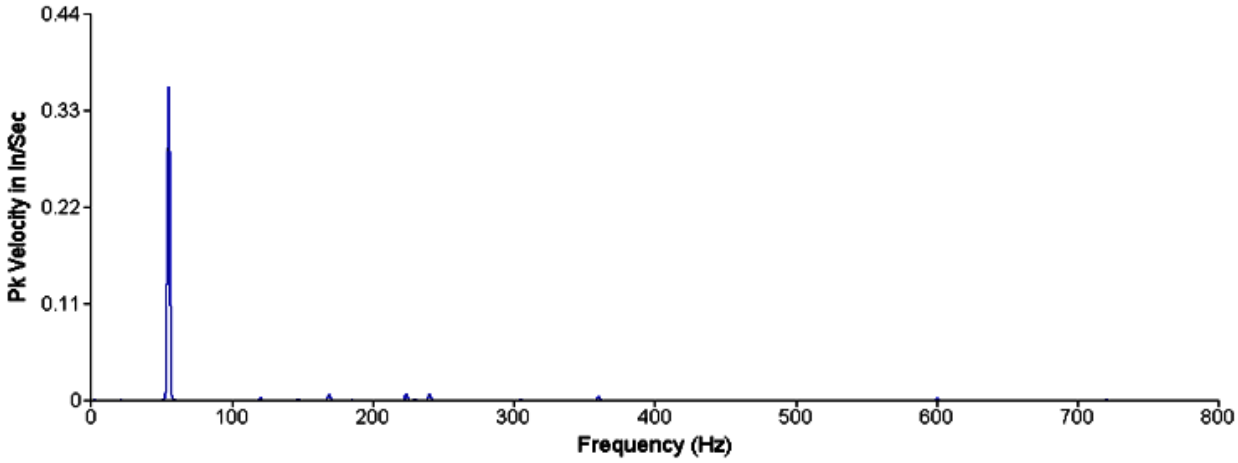
Step 1: Perform Integration

The waveform values are converted directly into velocity units before they are plotted to the waveform. Thus, the waveform is captured and stored in velocity.



Step 2: Perform FFT

The velocity waveform values are then used to perform FFT, resulting in a spectrum that uses velocity values for amplitude.



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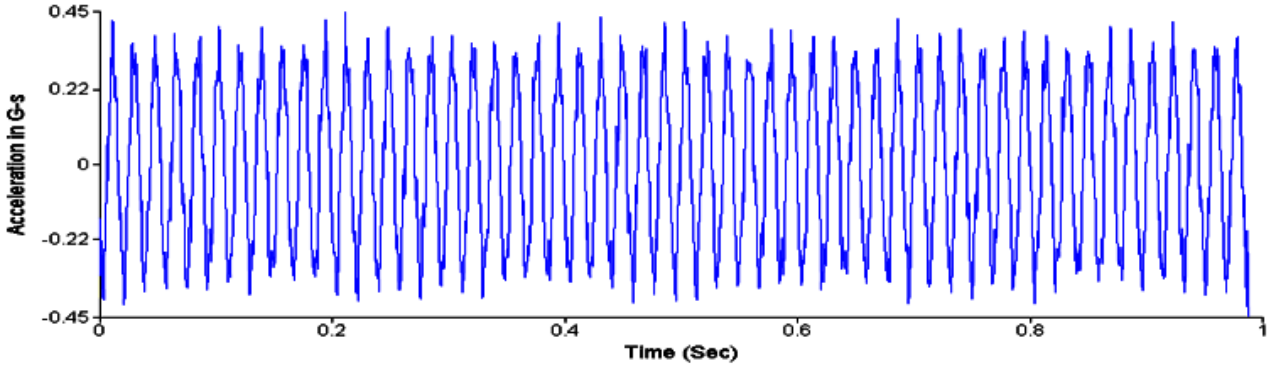
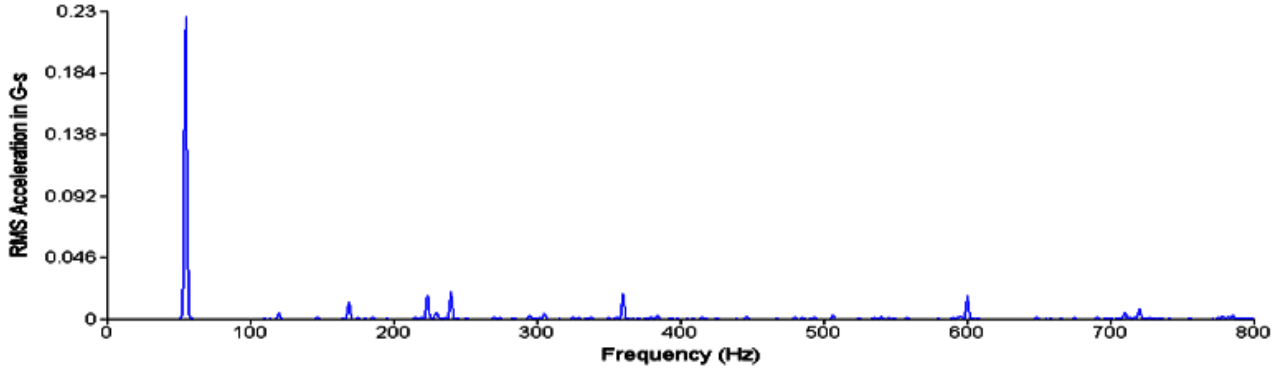
Digital Integration

Step 1: Use original acceleration waveform to perform FFT

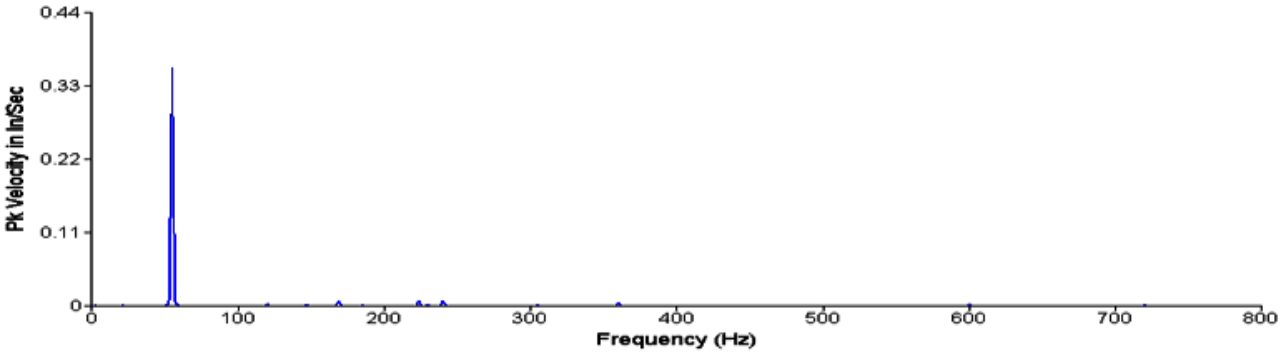
This method is often preferred by field analysts. It preserves the original time waveform in its native units.





Step 2: Perform Integration

Integration is then performed on the acceleration spectrum giving the user a spectrum with amplitude units in velocity.



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One drawback to integration is an amplification of the lower frequency data caused by the integration process. This effect is often called integration noise or a ski-slope effect. This is typically only noticeable when integrating from acceleration to displacement, and tends to affect only the lower one percent of the frequencies.

This integration error or ski slope may cause the overall vibration level to be higher than usual, since part of the ski slope may be included in the calculation of the overall vibration level. Normally, this is not of any real concern unless the analyst wants frequency analysis below 2 to 4 Hz.

Integration Pros and Cons

Digital Integration

Pro

- Easier to see impacting in the stored acceleration waveform.

Con

- Integration noise (ski slope) in low frequency spectrum.
- Needs software integration to view waveform in velocity.

Analog Integration

Pro

- Better low frequency spectral data.
- Better software integration and differentiation of the waveform.

Con

- Stored velocity waveform shows less impacting.

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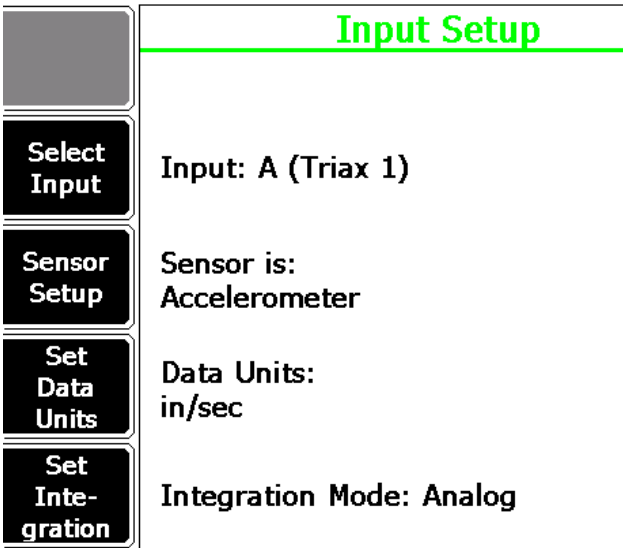
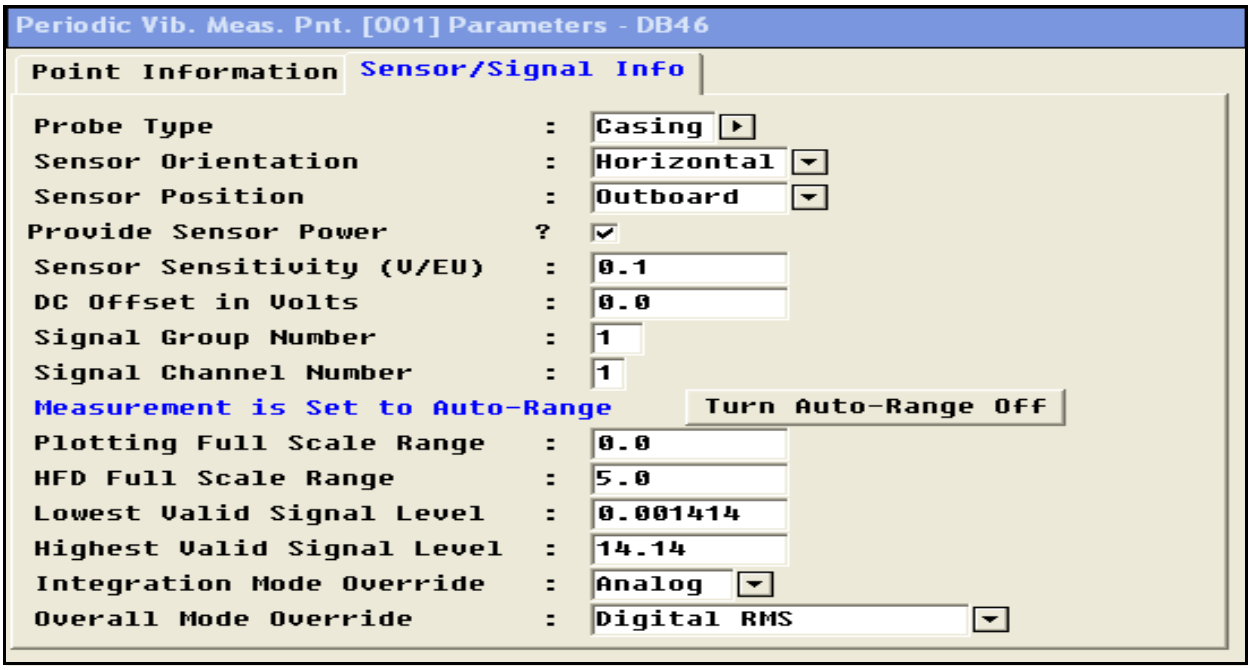
Integration in the CSI 2130



The integration mode is accessed in the CSI 2130 on the Input Setup screen. This setting needs to be considered any time the analyst is taking additional data in a route using Run Analyze, or any time the analyst is acquiring job data.



In the Route mode, an integration mode override is typically set up at the measurement point level. If the override is not set, then the analyzer's input setup is used to determine the type of integration. A popular method for integration control is to set analog integration at the measurement point level.

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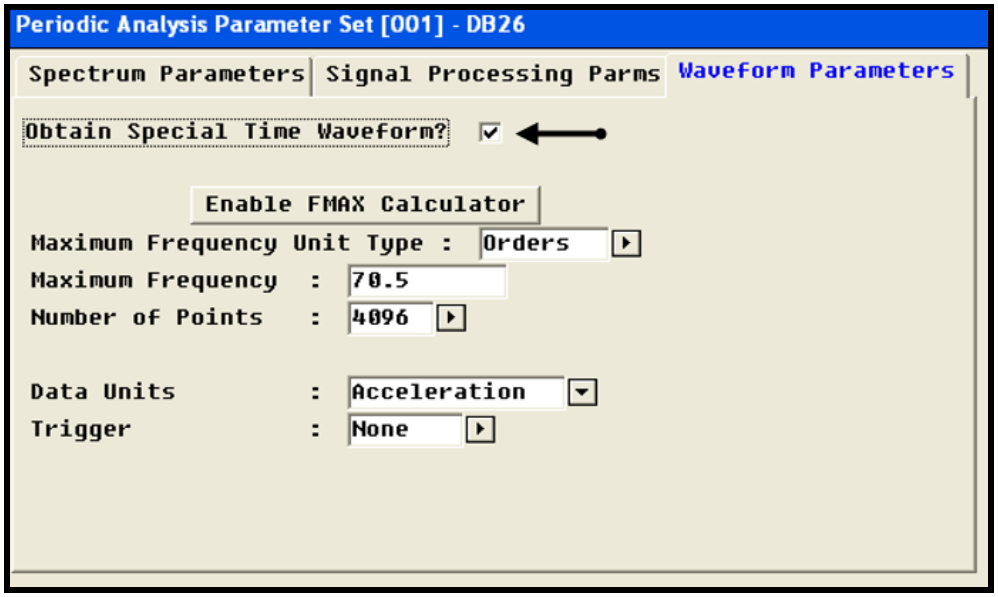
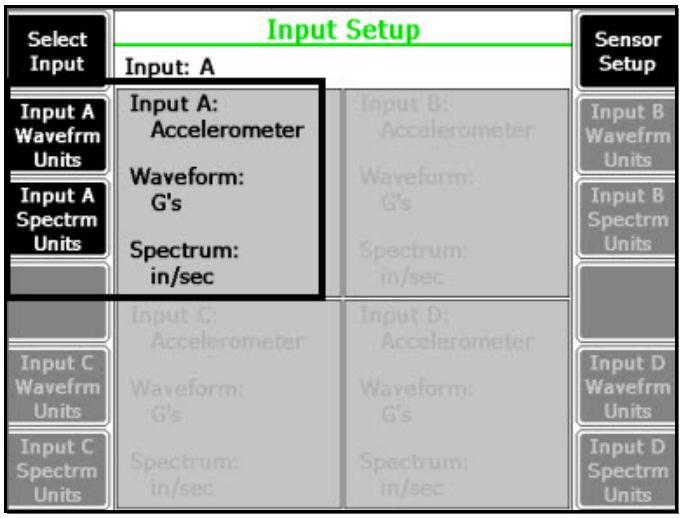
Then, use the **Obtain Special Time Waveform** option in the Analysis Parameter set. This allows the analyst to use analog integration for better spectral data, while storing an acceleration waveform to provide for better detail of any impacting that may be occurring.



The AMS 2140 uses analog integration. It integrates the waveform from acceleration to velocity as well as double integrating from acceleration to velocity to displacement. A Field Programmable Gate Array (FPGA) is used in the integration process. The fundamental principle is the same. However, the user selects integration in the time or frequency domains. In some instances, amplitude units are selected for the two domains.



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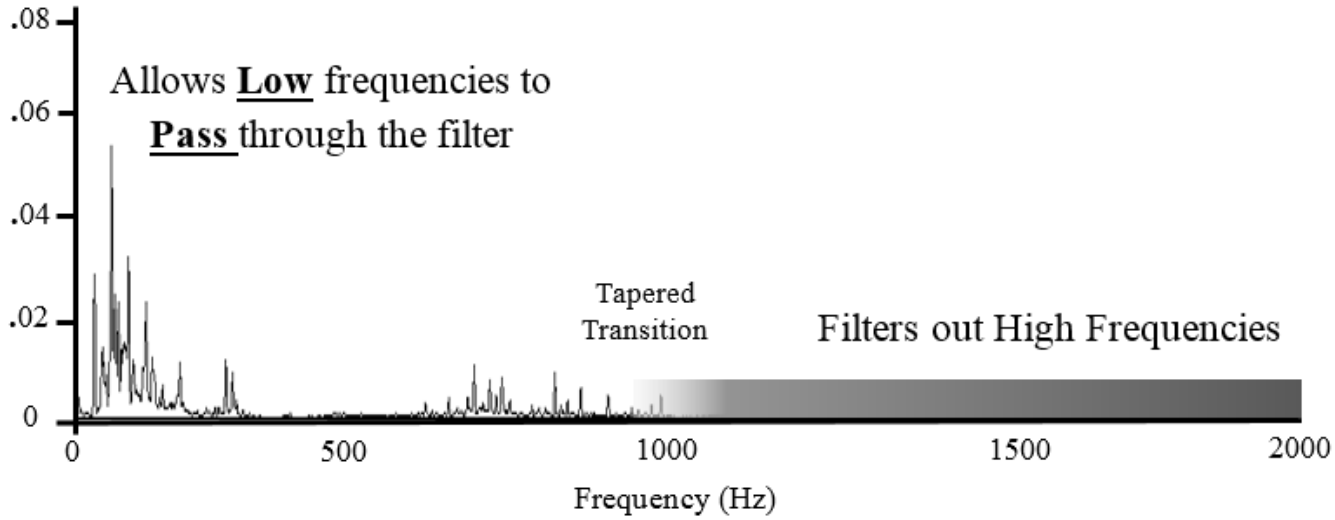
Filters

The vibration industry generally uses three types of filters:

- Low Pass
- Band Pass
- High Pass

Each one filters data out of a signal, which you may find useful when analyzing signals with large dynamic ranges. For example, some spectra have both large and small amplitudes relative to each other. Due to the dynamic range of the analyzer, you cannot analyze the low amplitude vibration in the same plot as the high amplitude vibration.

Low Pass Filter



As shown above, the **low pass filter** eliminates energy above the selected frequency. A low pass filter is what is used during normal data collection. It is set at the spectral Fmax to prevent aliasing.

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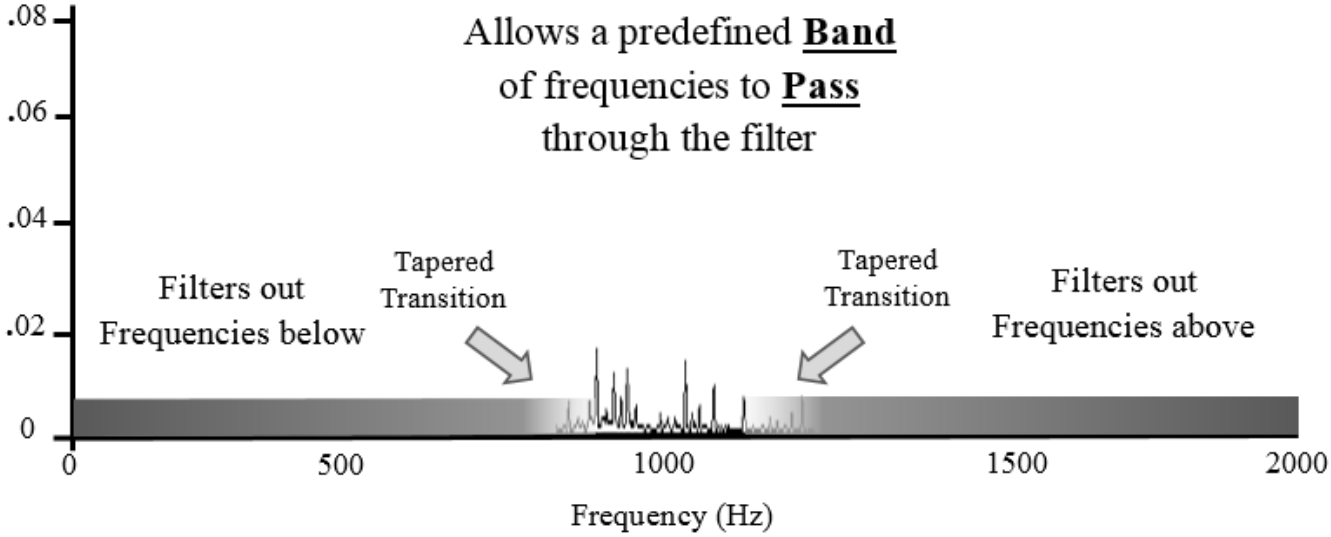
Band Pass Filter



The **band pass filter** filters out (eliminates) all vibration, both below and above the filter frequency. When the band pass filter is used, the user sets both a center frequency and a filter size. For instance, consider a shaft that turns 1800 RPM. A 1x band pass filter using a .1x RPM bandwidth would filter out everything below 1710 CPM and above 1890 CPM.

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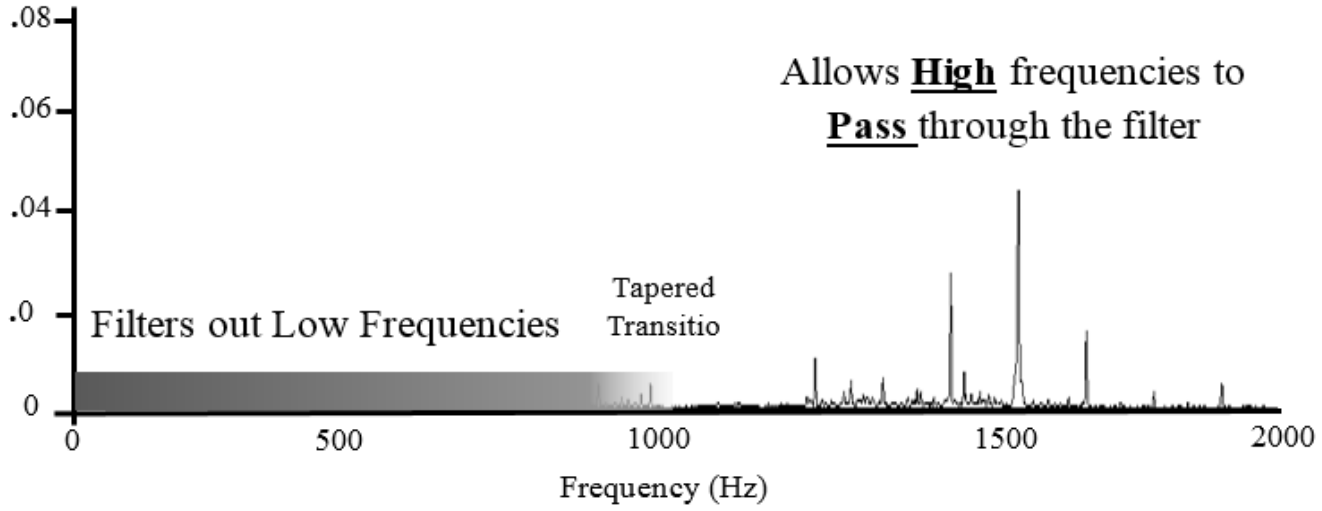
HighPass Filter



The **high pass filter** filters out (removes) everything below the filter setting. High pass filters are commonly used while acquiring PeakVue data, so that vibration caused by stress waves can be separated from vibration caused by other vibration forces.

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Review Questions

1. What would the digital sample rate be if a spectrum was acquired using a 2000 Hz Fmax and 1600 lines of resolution?

2. About how long would data acquisition take for a spectrum using a 1000 Hz Fmax, 1600 lines of resolution, six averages, with 67% overlap?

3. Assume that a Hanning window is being used in the previous example. What is the effective spectral resolution?

4. Spectral peaks are present at 119 Hz, and at 120 Hz. How many lines of resolution would be needed to accurately identify the frequency and amplitude of these vibration peaks if a 2000 Hz Fmax is used?

5. A spectrum is acquired using a 1000 Hz Fmax, a 2 Hz low cut off, 1600 lines of resolution, a Hanning window, and six averages. Which type of anti-aliasing filter is used, and what at which frequency is it set?

6. A machine is running 1780 RPM. The vibration spectrum acquired on that machine uses a 1000 Hz Fmax and 1600 lines of resolution. How many shaft revolutions will be captured in the time record?

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Phase Analysis

Section 3

Objectives

- Determine reasons for collecting peak and phase data.
- Describe and interpret peak and phase data in two formats.
- Discuss and demonstrate the phase relationship between acceleration, velocity and displacement.
- Perform class exercises using:
  - Peak and Phase
  - Cross Channel Phase

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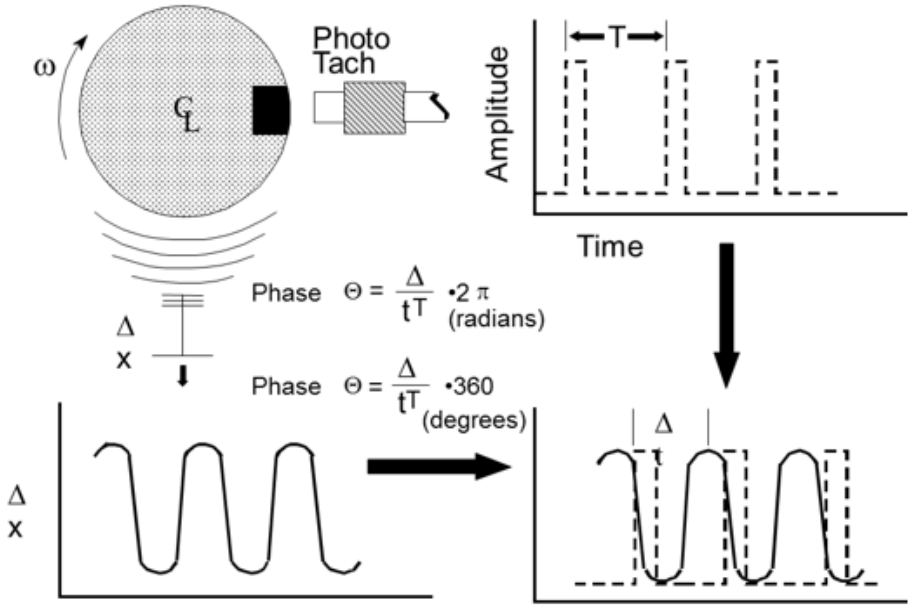
Phase Analysis

Phase Analysis is a troubleshooting tool that has many uses. One of the most common is used to determine the cause of a vibration that occurs at 1x RPM. Although imbalance is usually assumed to be the cause of vibration at this frequency, other problems such as misalignment, looseness, soft foot, bent shaft and resonance can also cause excessive vibration amplitude at 1 x RPM.

Phase can be defined as the relationship between two events. For vibration diagnostic purposes, it is classified two ways.

True Phase

The first classification is **True Phase** (also referred to as Absolute or Synchronous Phase). True Phase refers to the location in degrees (or radians) that marks a machine's vibratory high spot and its frequency with respect to a fixed reference mark (reflective tape) on the rotating component (shaft). True phase is measured only at 1xTS or any harmonic up to 128 orders. The defining characteristic of true phase is that it is measured by referencing a fixed point on the rotor. This is usually either a piece of reflective tape (if using a photo-tach), or a key or keyway, (if using an eddy-current sensor). When measured with Emerson equipment, the measurement is made beginning with the reference pulse and ending with the vibratory high spot in the waveform. This method of measuring true phase is sometimes referred to as Reverse Phase.



Single Channel Phase Measurement (@1XTS)

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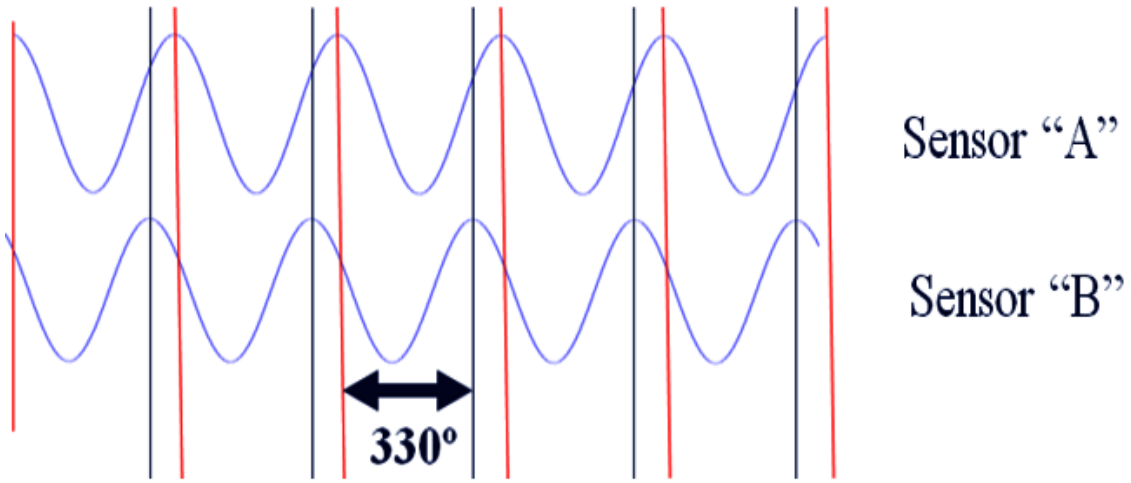
Relative Phase

The second classification, **Relative Phase** (also referred to as Cross Channel Phase), refers to the relationship in degrees (or radians) of the vibratory high spot in the waveform of one sensor compared to the vibratory high spot of a second sensor acquired at exactly the same time at exactly the same frequency.

Unlike True Phase, Relative Phase can be measured at any frequency, regardless of whether it is synchronous or not. Also, unlike true phase, relative phase measurements do not reference any recognizable spot on the rotor, such as reflective tape or a keyway. Therefore, it is not suitable for use with balancing applications.



Either type of phase can be used in most diagnostic situations. Regardless of which type of phase is measured, the results are commonly documented on a Phase Analysis Bubble Diagram, an example of which is shown below.

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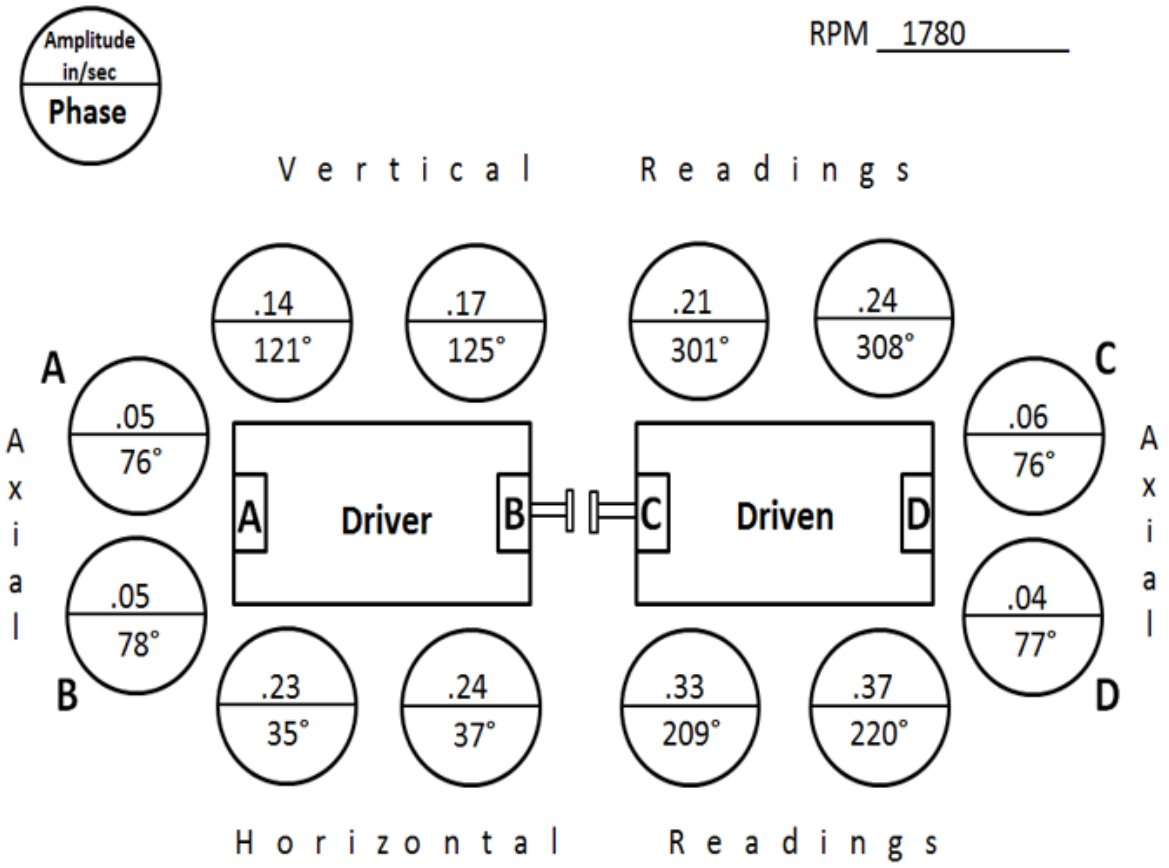

When analyzing phase analysis data, both the amplitudes and phase values need to be considered. In this example, it appears that the vibration amplitudes of the driven equipment are marginally higher than those of the driver. Also, the amplitudes in the horizontal direction seem to be slightly higher than those taken from the vertical or axial directions. By looking at the phase characteristics, it would appear that there is about a 180 degree phase shift across the coupling.

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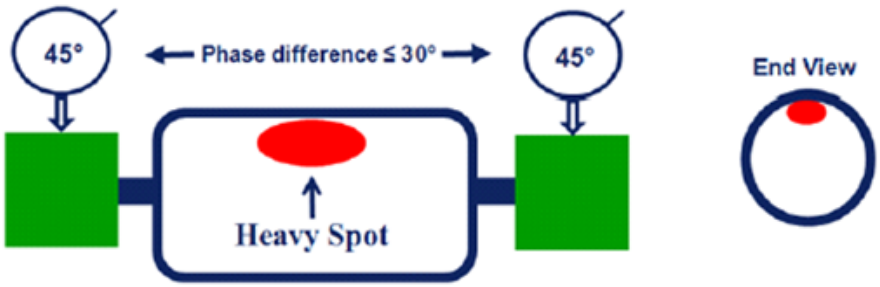
Phase Characteristics of Common Vibration Problems

Imbalance

Imbalance refers to the condition of a rotor whose center of mass is not the same as the center of rotation. In most diagnostic situations, three types of imbalance are recognized.

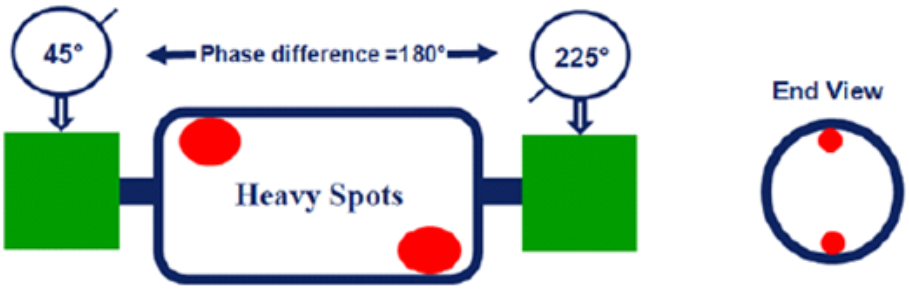
Static Imbalance

With static imbalance, the readings will appear as a zero degree shift across the rotor radially. Moving the sensor from horizontal to the vertical direction will result in a difference of 90 degree on the same bearing location. The excessive mass resides at a single angle across the width, or on different planes on the rotor.



Couple Imbalance

Couple imbalance is a case of imbalance where the masses are of equal value and separated by 180 degrees.



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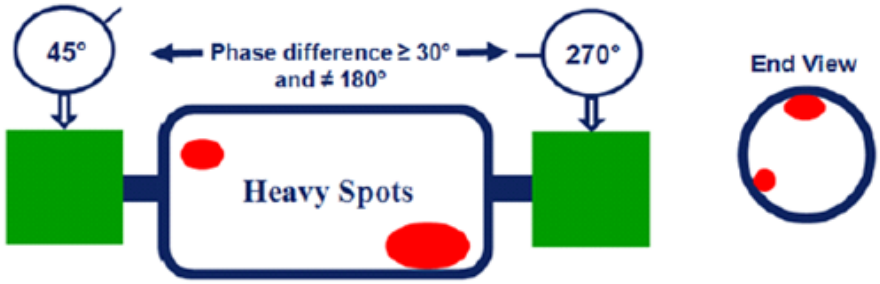
Dynamic Imbalance

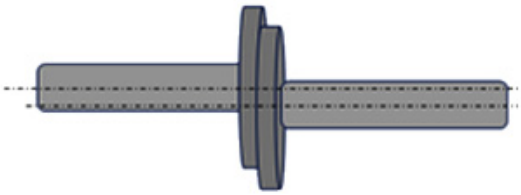
Dynamic imbalance tends to appear as a shift across the rotor radially related to the heavy spots on each end of the rotor. If the heavy spots are 180 degrees opposite of each other, the phase readings will usually be 180 degrees out of phase.



Misalignment

Misalignment refers to the states of two coupled shafts that are either not perfectly parallel, or do not share a common axis.

Parallel or **Offset** misalignment tends to cause a 180 degree phase shift across the coupling, when measured in the radial direction.



Angular misalignment tends to cause a 180 degree phase shift across the coupling, when measured in the axial direction.

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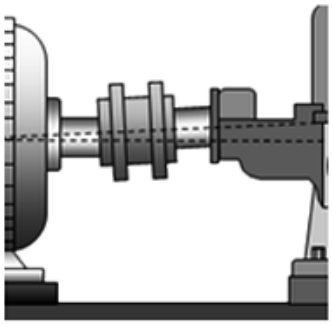
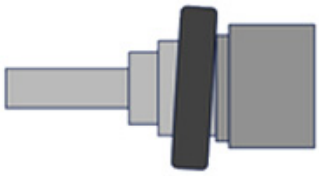


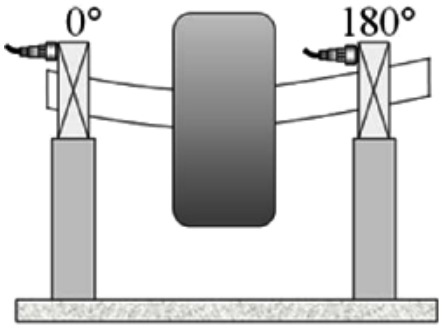
Bearing misalignment (cocked bearing) tends to cause a 180 degree phase shift across the face of the housing. Using an analog clock face as an illustration, 12:00 would be 0 degrees, 6:00 would be 180 degrees, 3:00 would be 0 degrees and 9:00 would be 180 degrees, relatively.



Bent Shaft

In a machine with a bent shaft, the vibration response depends on the location of the bend.

A shaft bent near the center tends to show up as 180 degree phase shift across the bend, not the coupling.



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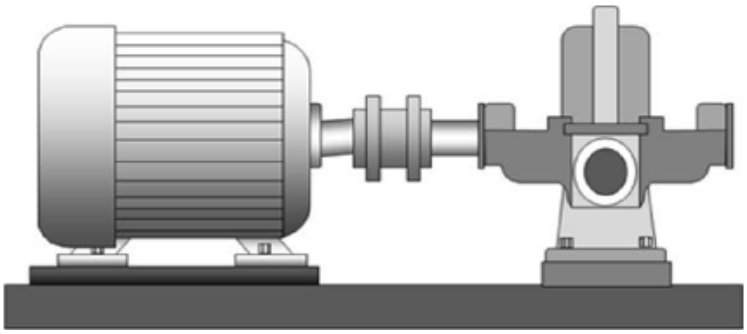
If the bend is at the end, past the bearing, the vibration tends to appear as a high 2 x TS peak, with a phase reading of 180 degrees across the face of the machine.



To prove bent shaft, shut the unit down, break the coupling and mount a dial indicator on the suspect shaft. Rotate the shaft and watch for excessive deflection.

Looseness

Looseness can originate from several sources. The foundation, sole plate, floor, grout and other laminate components of a machine may be compromised so that there is no longer good contact between them. Occasionally, an alignment shim may work its way out of the structure or the grout may crack. The looseness can also be internal such as a bearing's inner race loose on the shaft or the outer race loose in the housing.

When looseness is the dominant cause of vibration, the phase measurements are very erratic. Often, phase measurements are so erratic that phase cannot be accurately measured.

Resonance

Resonance is defined as the excitation of a natural frequency by some forcing function. Occasionally, it is the vibration force at the RPM of the rotor that excites the rotor's natural frequency. The RPM at which this happens is referred to as the machine's critical speed. The phase characteristics of a machine running above its critical speed are different than that of a machine running at, or below its critical speed. When coasting down from above the critical speed, the machine will experience a 90 degree phase shift when it reaches its critical speed, and then another 90 degree shift as its RPM continues to drop to well below its critical speed. Overall, there will be a 180 degree phase shift during the entire process.

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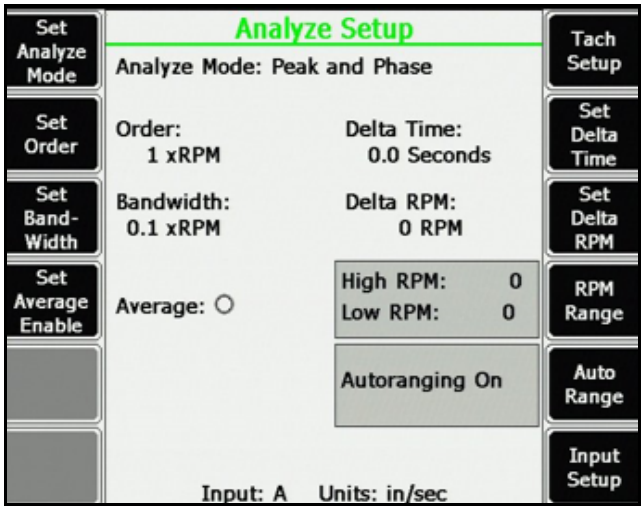
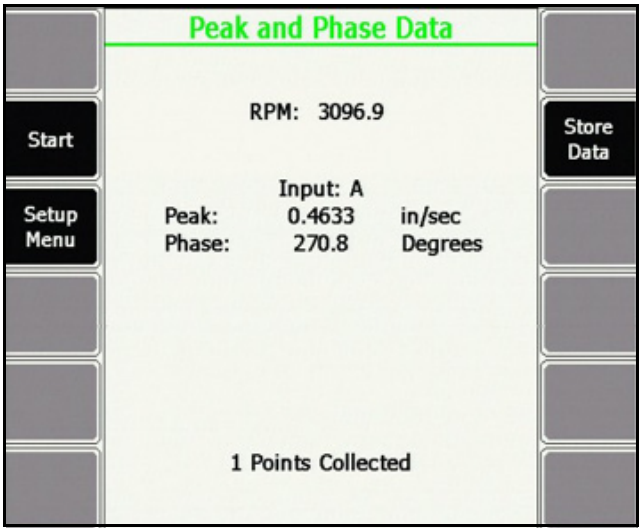
Measuring Peak and Phase Values for Diagnostic Phase Analysis

Single Channel: True Phase

When using the analyzer to measure true phase, the **Peak and Phase** application is used.



Since the Peak and Phase application measures true phase, a photo-tach is needed along with reflective tape on the shaft of the machine being analyzed. Place a single sensor at one of the measurement locations on the machine and acquire the data.



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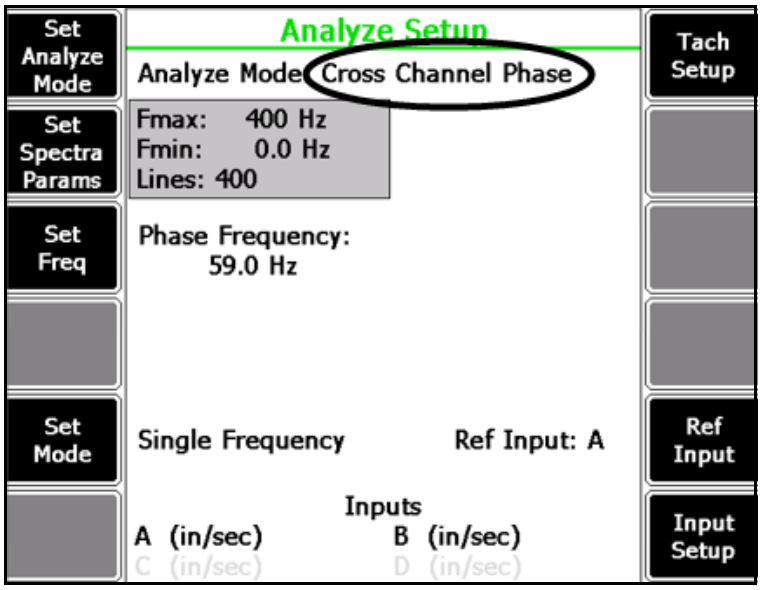
Enter the Peak and Phase values into the bubble diagram for that location. Next, move the sensor to another measurement location on the machine and repeat Peak and Phase data acquisition until all accessible measurements have been completed and documented. An analysis of the bubble diagram is then performed in order to identify the cause and severity of any suspected problems.

Cross Channel: Relative Phase

When using the analyzer to measure relative phase, the **Cross Channel Phase** application is used.



Since no tach pulse is used, it is extremely important that the **Phase Frequency** be set to within one bin of resolution of the actual rotor RPM. Also, the amplitude units should be the same for each sensor.

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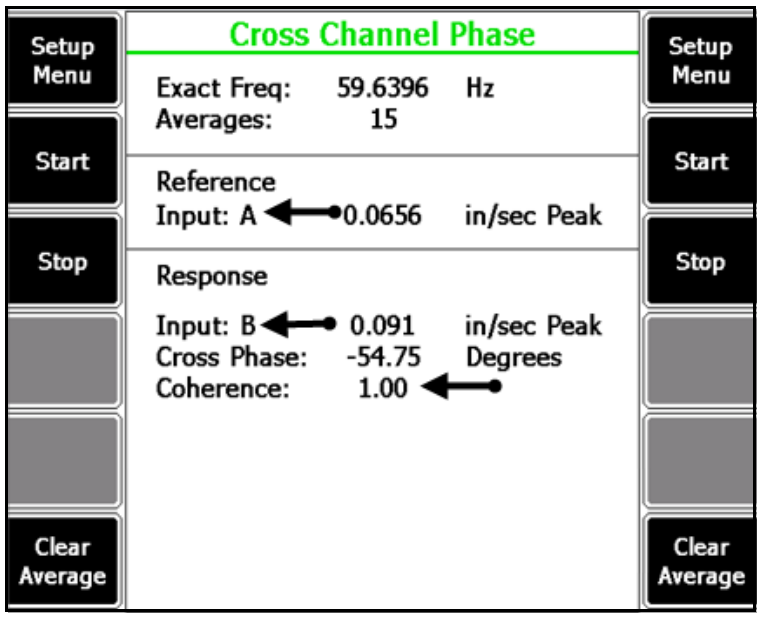
When Cross Channel Phase is used, one of the signals is used only for reference. In this example the reference signal has been designated as that from sensor A. If sensor A is placed on one of the measurement locations of the machine, then the values for that position on the bubble chart will be .065 in/sec at 0 degrees. The values for the position of sensor B will be .09 in/sec at -55 degrees. The remaining measurements are made by moving only sensor B to the remaining measurement locations. The reference sensor, (A) needs to remain in the same location for each of the measurements, just as the photo-tach is not moved when acquiring data using the single-channel method.



It should be noted that averaging is used with cross channel phase and because of this, the averages need to be cleared every time the sensor is moved to a new location.

The **Coherence** value should be at least .9 in order for the data to be considered valid. Coherence is measured on a scale of zero to one, with a value of one indicating perfect coherence. A low coherence value indicates that the two sensors are not measuring vibration energy from exactly the same source.

Note

A low coherence may indicate extreme looseness between a foot and its base. The data is still valid.

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Start - Up Coast Down Test

The **start – up / coast down test** is commonly used to identify a machine's natural frequency or its critical speed. In order to allow for enough sample time, the machine should coast down gradually. If it comes to a stop too rapidly, the analyzer will not be able to acquire enough data for a valid analysis.

To perform a coast down test, begin monitoring **Peak and Phase** at the desired location(s), while the machine is running at full speed. Although Peak and Phase is a monitor mode application when used to acquire data for diagnostic phase analysis, if a speed change occurs during the analysis, the application automatically begins to store and trend the changing values until the speed stops changing. While monitoring the peak and phase data with the analyzer, secure the power to the machine. The analyzer will continue to automatically measure and store the changing peak and phase values until the machine coasts down to a stop, at which time the analyzer will automatically stop acquiring data. Once the analyzer screen reads, **tachometer timeout**, the data can be stored and analyzed.

If the machine is running at resonance, there should be an immediate reduction in vibration amplitude, along with a 90 degree phase shift when the machine is shut off.

If the machine is operating below the system natural frequency, the amplitude should decrease in a linear fashion as the rotor coasts to a stop. There should be, at most, a very minimal phase shift.

If the machine is running above either its critical speed or the system natural frequency, the amplitude will decrease gradually until it approaches the natural frequency. The amplitude will increase exponentially and the phase will shift 90 degrees. Then, as the RPM decreases further to well below the natural frequency, the vibration amplitude will decrease and the phase will shift another 90 degrees.

The phase difference from full RPM compared to the phase reading after the rotor coasts down below the system natural frequency or critical speed will be a total of 180 degrees.

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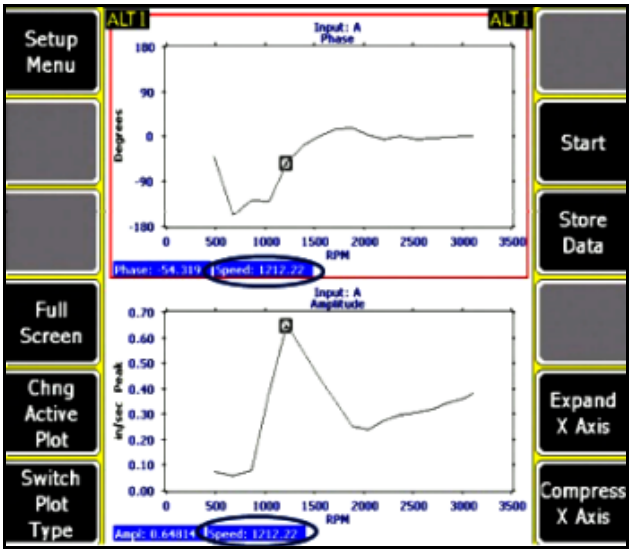
Interpreting Coast-Down Results

Coast down data acquired using the Peak and Phase application can be viewed as Bodè plots of Amplitude vs RPM and Phase vs RPM, or as a single Nyquist plot. If there is a significant amplitude increase at a certain RPM, but little phase change at that frequency, it suggests that a system natural frequency was excited at that frequency. If the amplitude increase is accompanied by a significant shift in phase at that frequency, then it suggests that this is a rotor critical speed.



In this example, the phase experienced a shift of approximately 180 degrees, with a 90 degree shift at the same 1212 RPM as the maximum amplitude (peak) occurred. Viewing both plots helps to identify not only the natural frequency near 1212 CPM, but also the probability that this is a rotor critical at 1212 RPM due to the significant phase shift at that same frequency. If the phase shift does not occur, then it is more likely that the amplitude increase is due to the excitation of a system natural frequency instead of a rotor critical speed.

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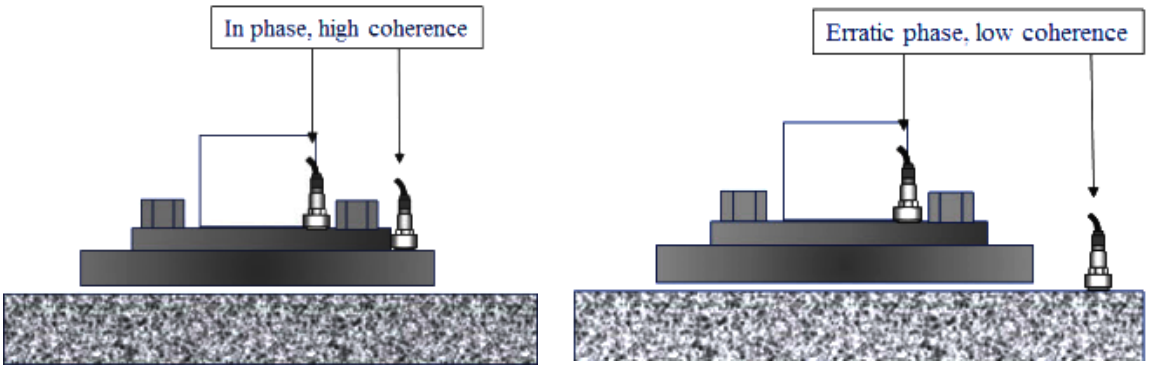
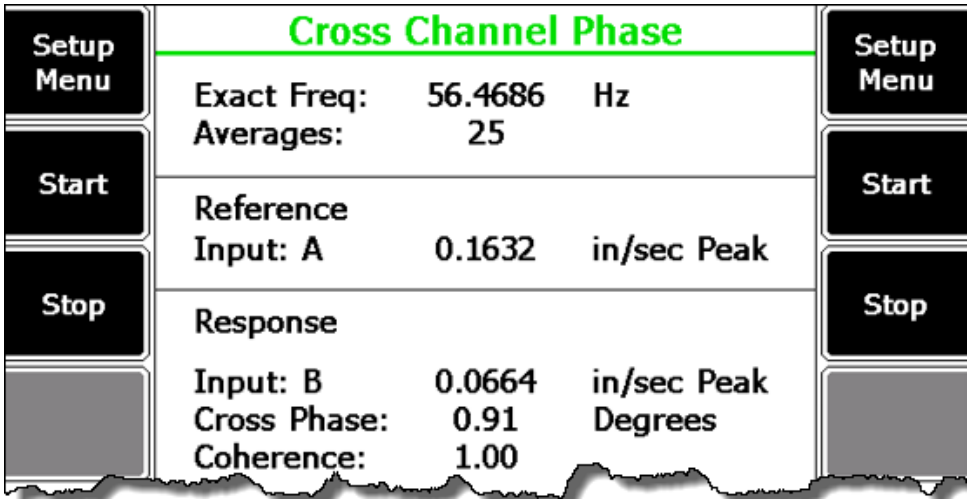
Using Phase Analysis to Identify a Loose Joint

Either type of phase can be used to help identify the location of a loose joint. The advantage of using cross channel phase is that the coherence value can also be used in some cases to help identify the separation.

In this illustration, cross channel phase is being used. The reference sensor is placed in the vertical direction on the machine foot. The other sensor is mounted to the base plate. A second measurement is then made by moving the second sensor to the floor.



In the first data, the relative phase is very close to zero, and the coherence is perfect.



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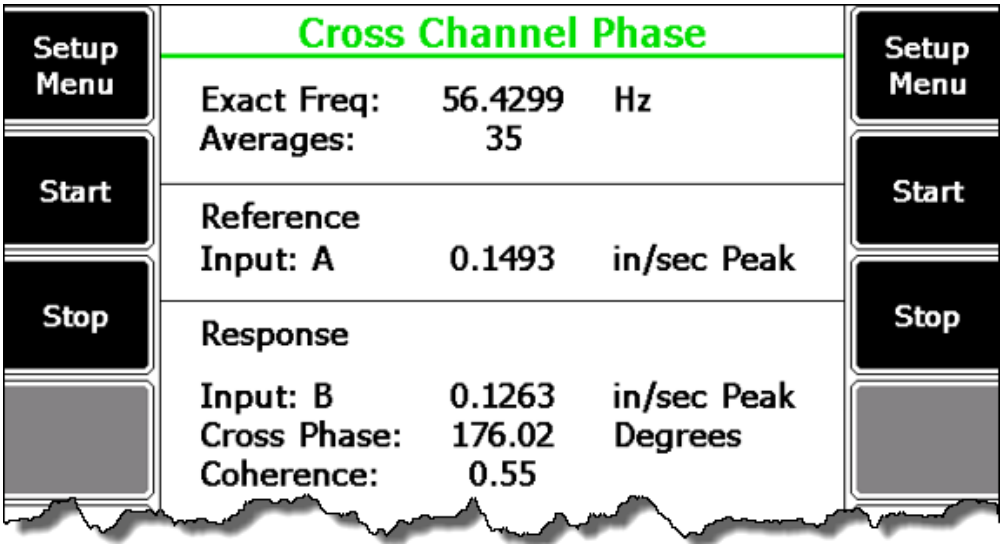
In the second data, the phase between the two sensors is nearly 180 degrees, and the coherence is much lower. This is a good indication that the loose joint is located between the machinery base plate and the floor.



Class Exercise

- Perform a coast down using the Peak and Phase application.
- Store and analyze Bodė and Nyquist plots.
- Perform a phase analysis on the classroom rotor kit using the Cross Channel Phase application and a bubble diagram.

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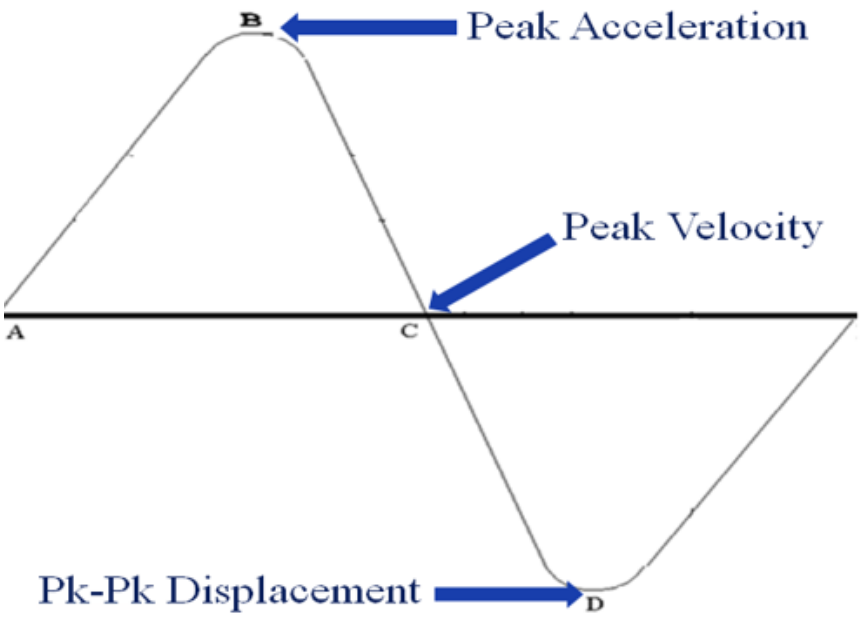
Effects of Amplitude and Phase

When phase is being measured, either for diagnostic purposes or for balancing, it should be understood that different amplitude units have different phase relationships. A peak and phase measurement that uses amplitude units in displacement will have a different phase value compared to a measurement taken at the same location using amplitude units in velocity or acceleration. It is important that the amplitude units remain consistent while acquiring peak and phase data. Also, when acquiring cross channel phase, the sensors need to be of the same type, and the amplitude units for each sensor need to be the same.

In order to better understand the phase relationship between different amplitude units, consider a simple sine wave and relate the signal to a vertically mounted accelerometer. As the sensor reaches the upward limit of travel it slows down, and for a very brief instant as it changes its direction of travel from up to down, it stops. This is the point marked B in the illustration. Because there is a direction change, this is the point where the acceleration, or change of velocity, is the greatest. This is the point of peak acceleration.



However, the measured velocity at this point is zero. Velocity continues to increase from this point until it reaches halfway to the opposite limit of travel, which is point C. As it passes this point, it will begin to slow down, because it will once again stop at point D. The velocity is greatest at point C, which is 90 degrees of shaft rotation after point B.

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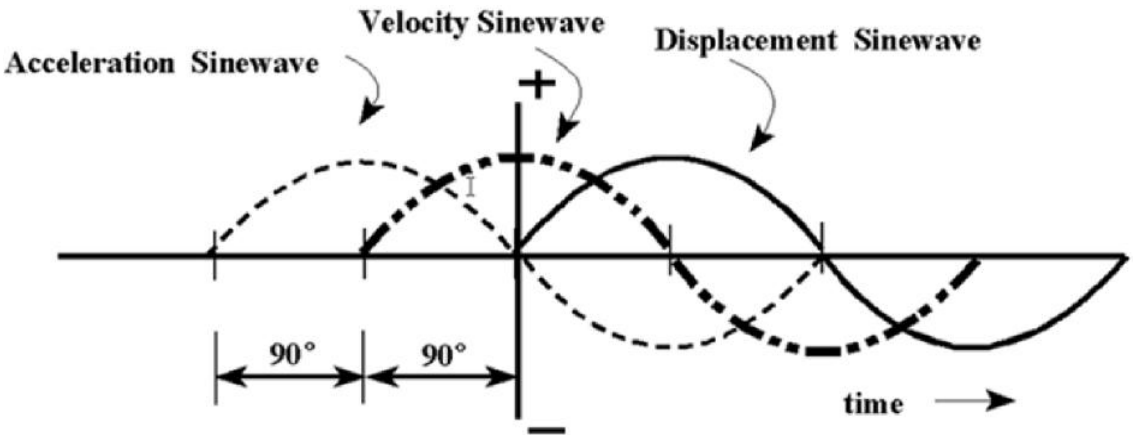
The overall distance traveled, or peak-to-peak displacement, cannot be measured until the entire distance is completed, which is at point D. The displacement value is not measured until 90 degrees of shaft rotation after the measurement of peak velocity.

This is the reason acceleration leads velocity by 90 degrees, and velocity leads displacement by 90 degrees, and acceleration leads displacement by 180 degrees. The figure below illustrates how the sine waves from acceleration, velocity, and displacement relate to each other in phase.



It should be understood that when peak and phase measurements are made with the analyzer, the integration used to convert acceleration values to velocity values affects the phase values as well. When using the peak and phase application, if the amplitude units are velocity, then the phase value represents the velocity sine wave. If the amplitude units are changed to displacement, the phase value will also change to reflect the displacement sine wave.

Class Exercise

- Use the Peak and Phase application to measure amplitude and phase at one measurement point on the class rotor kit using acceleration, velocity and displacement amplitude units.

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Summary

Phase Analysis is a useful tool for identifying the cause of vibration at 1X RPM.

Spectral and waveform patterns may be similar for different forcing functions. Phase analysis may then provide the additional information needed to more accurately identify the true nature of the forcing function. Occasionally, phase data is the only way to discern one fault from another thereby suggesting the appropriate resolution.

The two types of phase measurement are True Phase and Relative Phase.

The phase relationships throughout the machine and between sections of the machine aid the analyst in visualizing the movement of the machine and diagnosing the root cause of the vibration.

The Peak and Phase application is an effective way to perform a coast down test for identifying a machine's natural frequencies and critical speeds.

The data acquired from phase analysis is often documented and analyzed using a Bubble Diagram. Both the amplitude and phase need to be scrutinized during the analysis.

Acceleration, Velocity and Displacement all have different phase relationships.

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Review Questions

1. What frequency of vibration is usually of interest when measuring Peak and Phase?

2. True or False. A True Phase measurement requires the use of two vibration sensors.

3. True or False. Cross Channel Phase can be used to measure phase at frequencies other than 1 x RPM.

4. When measuring Relative Phase, what is the lowest acceptable coherence value?

5. Name two data plots that can be viewed and stored after completing a Peak and Phase coast down test.

6. By how many degrees does velocity lead displacement?

7. If a machine is operating above its first critical speed, the phase angle could shift as much as _____ during a coast down test.

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8. Relative Phase is also referred to as _____ phase.

9. As a rotor coasts down through its critical speed, the amplitude increases. In addition to the increase, at the same RPM the phase will shift _____ degrees.

10. The phase lag between acceleration and displacement is _____ degrees.

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Imbalance

Section 4

Objectives

- Define imbalance.
- Explain various types of imbalance.
- Describe vibratory characteristics of imbalance.

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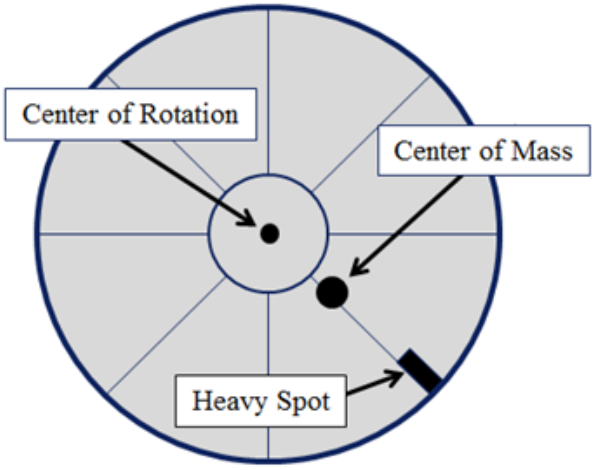
Imbalance

Imbalance is defined as the state or condition of a rotating body whose center of rotation is not its center of gravity or center of mass. This condition results in a heavy spot on the rotor.



The heavy spot produces a centrifugal force that causes the body to rotate off the center, which causes increased vibration at the rotational frequency.

This force generates a sinusoidal waveform, occurring once per revolution of the shaft.

The spectrum will show a single peak at turning speed.

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Imbalance and Phase

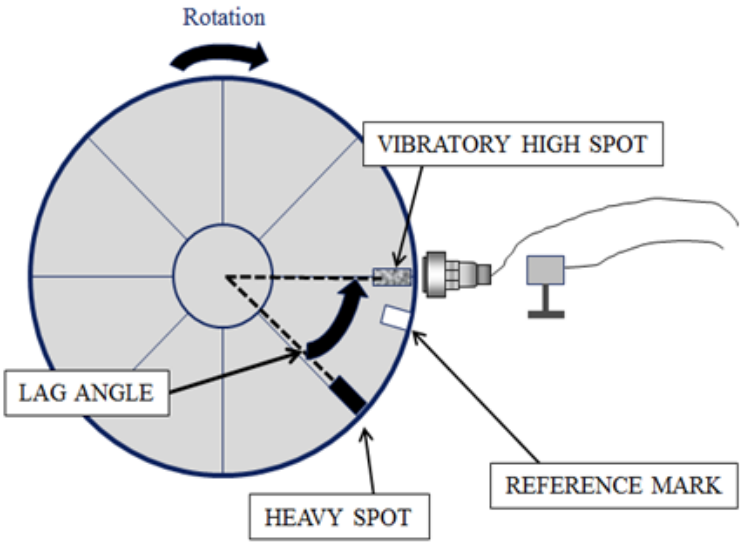
It is important to understand that a vibration sensor and a tachometer do not directly identify the heavy spot on the rotor. The **vibratory high spot** is actually identified.

The vibratory high spot is defined as the measured response to the heavy spot.

This is the point where the measured turning speed vibration amplitude is highest.

Phase Lag

A **phase lag** exists between the actual heavy spot on the rotor and the measured high spot. This lag is a product of the mass, stiffness and damping properties of the system. Consider an imbalance force rotating at 1800 RPM with the sensor mounted on a massive machine. This mass will resist moving as the force rotates past the sensor. Some amount of time elapses between the force passing the sensor and the mass of the machine responding to the force. By the time the sensor actually responds to the force, the heavy spot has moved past the sensor. This time difference is referred to as **phase** or **system lag**.



Heavy Spot: Centrifugal force from imbalance.

Vibratory High Spot: Response of the force.

Lag Angle: Angle between the heavy spot and the vibratory high spot.

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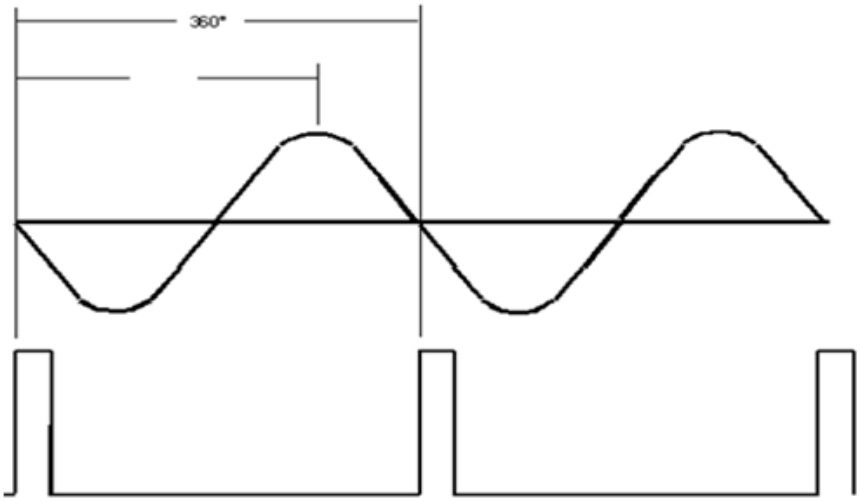

One of the parameters that affect the phase lag angle is the relationship between the RPM and the rotors' critical speeds.

When RPM is:

Below critical speed the lag angle is close to 0 degrees.

At or near critical speed the lag angle approaches 90 degrees.

Above critical speed the lag angle will be above 90 degrees.

Note

Vector balancing uses a known weight placed at a known location on the rotor to determine the location of the heavy spot on the rotor. The actual lag angle is calculated from the results of the trial run.

	Dominant Frequency	**Dominant Plane**	**Phase Reading**
Static	1x TS	Radial	Radial In Phase
Dynamic	1x TS	Radial	Radial 0 – 180 out / 2 plane required
Couple	1x TS	Radial / Axial	Radial Out of phase
Overhung Rotor	1x TS	Radial / Axial	Radial unsteady, Axial in phase

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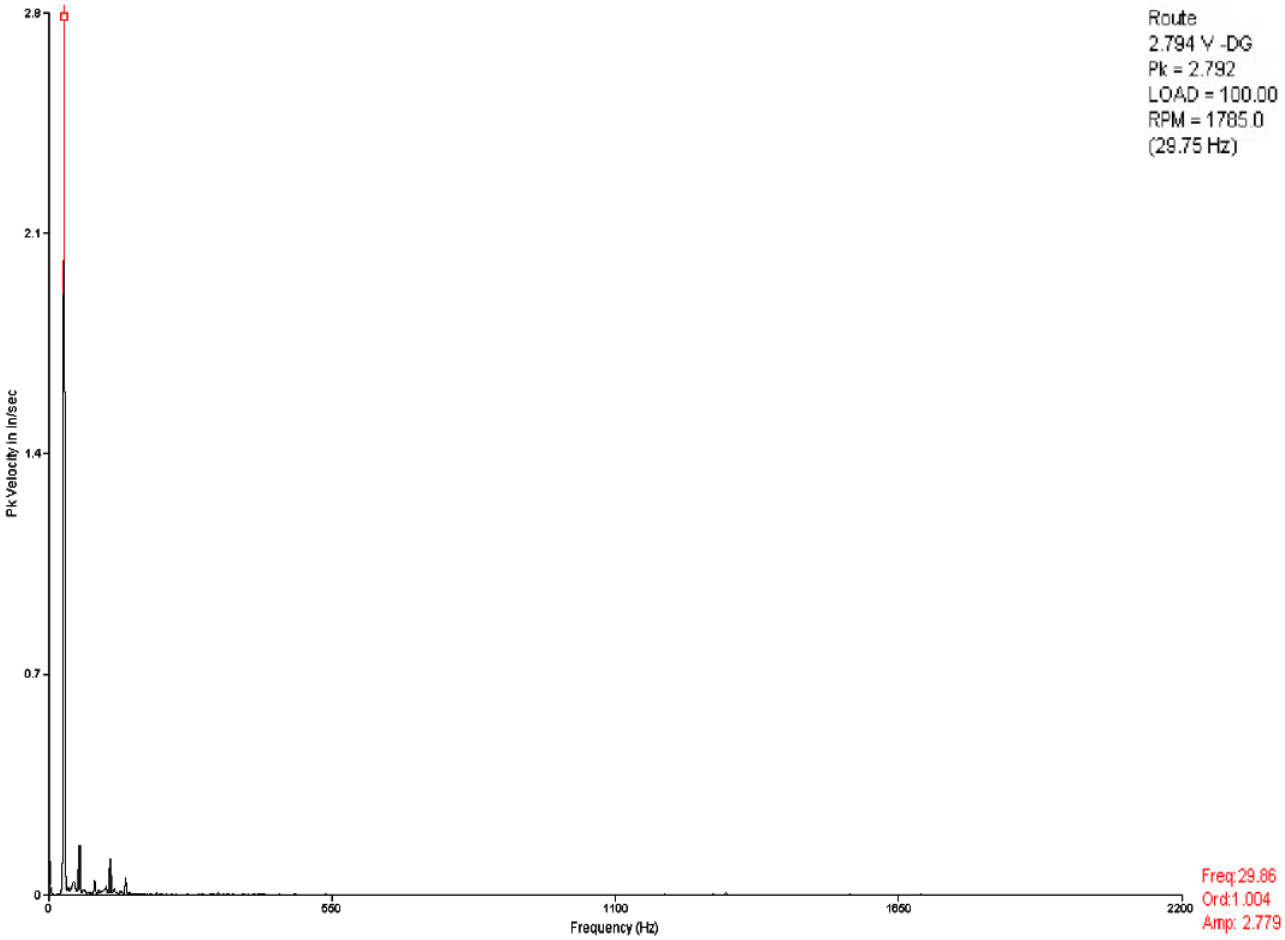
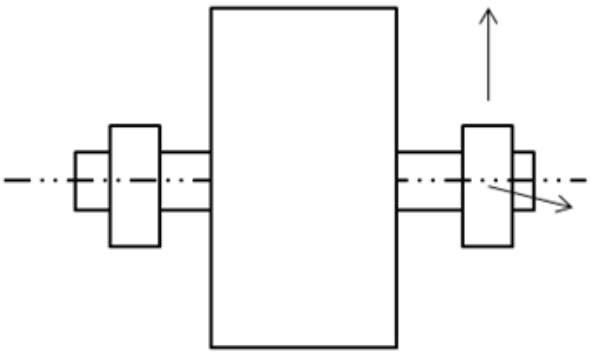
Imbalance – Spectral Data

When imbalance is the primary forcing function, a peak at 1 x TS will appear in the spectrum. Harmonics of 1 x TS will be relatively low in amplitude.



Imbalance force is generally in the radial direction for center-hung rotors.



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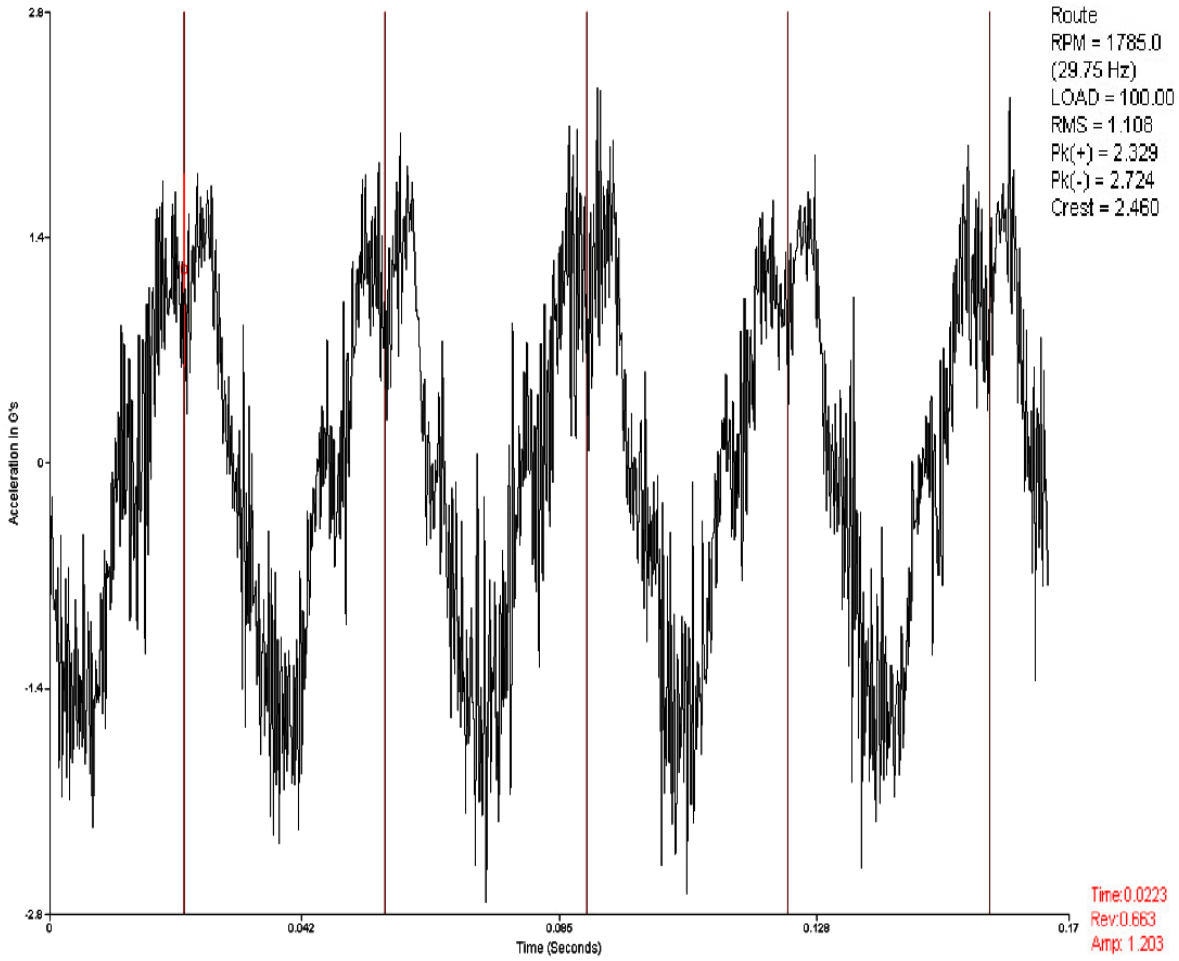
Imbalanced over-hung rotors normally generate a high amplitude at 1 x RPM in the radial direction. Occasionally, this amplitude is high in the axial direction as well.

When imbalance is the primary forcing function, the time waveform will have a periodic, sinusoidal pattern. The period in the waveform will be equal to the inverse of the rotational frequency of the shaft.

Imbalance generates the most sinusoidal pattern compared to all other defects. The amplitude units will affect the appearance of the waveform.

Changing the amplitude units can help determine the pattern(s) of the waveform.

Acceleration



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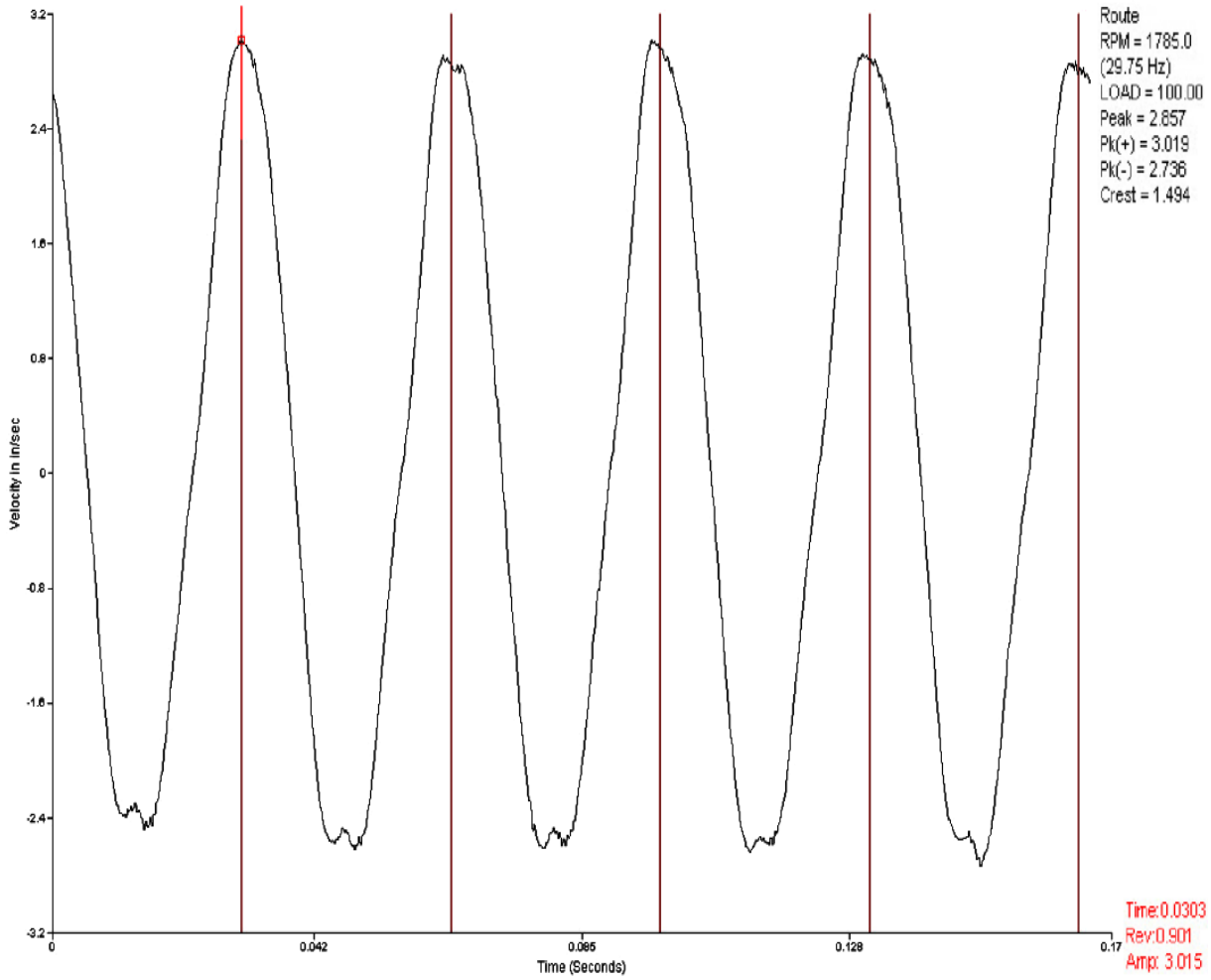
Velocity



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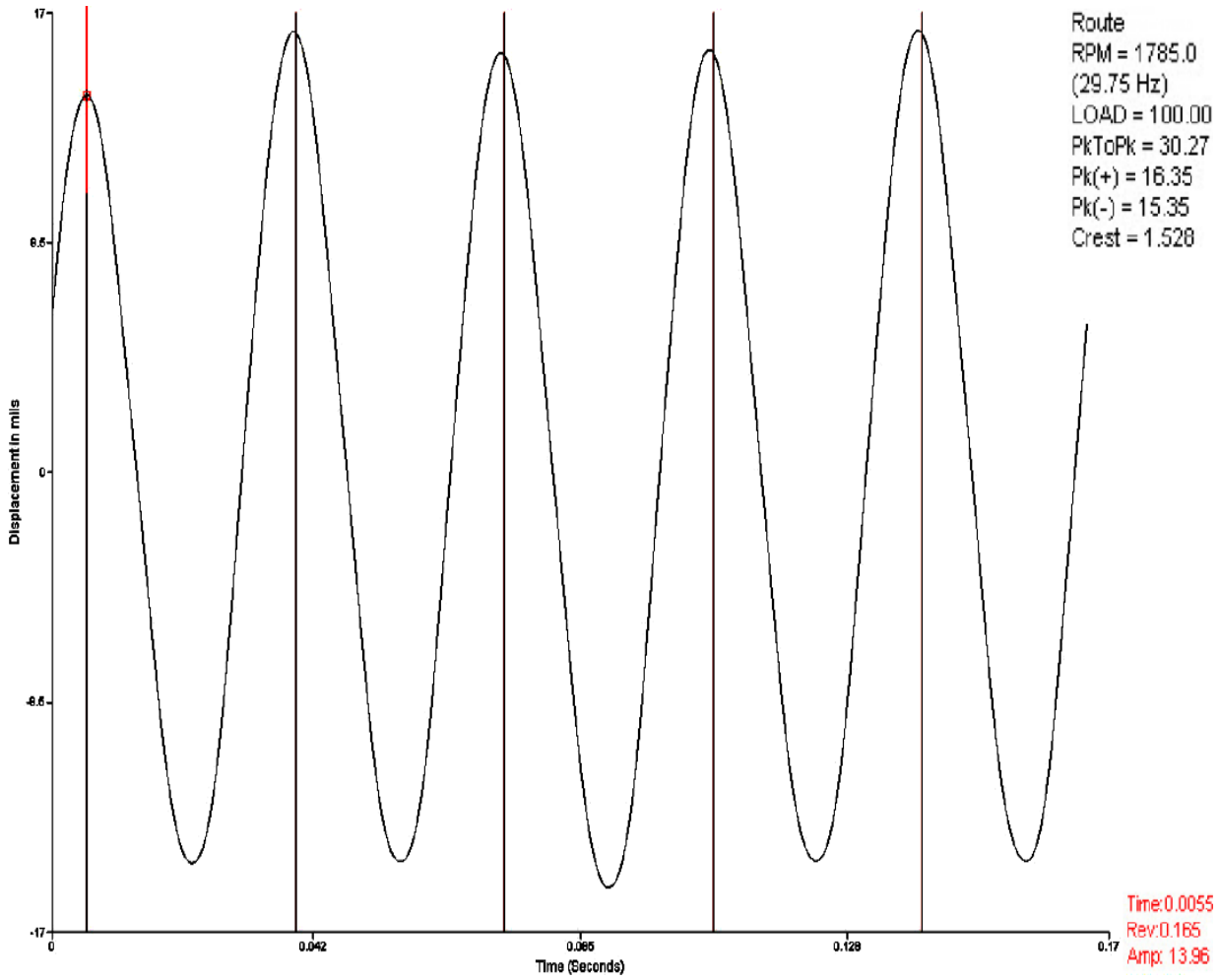
Displacement



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Imbalance Force

To better appreciate the relationship between rotational frequency and the forces involved, it is important to understand the fundamental equation for calculating the force due to mass imbalance.

As the speed of the rotor is changed, the effect of the change is not linear. The response is exponential due to omega ω^2.

Determine the force imbalance:

$$\mathbf{F_T = \omega^2 \times UB/ g}$$

Where:

$\mathbf{F_T}$ = Force due to mass imbalance

$\boldsymbol{\omega}$ = shaft turning speed (RPM)

$\mathbf{U_B}$ = mass x eccentricity (m x e)

m = mass

e = eccentricity (distance, in inches, of the mass from the center of the rotor)

g = gravitational force (386.4 in/sec^2)

Omitting the constant, force of gravity, the equation is simpler and the results will be easy to follow.

$$\mathbf{F = \omega^2 \times m \times e}$$

Speed change factor	Mass	Eccentricity	Speed	**Force from unbalance**	**Square of speed change factor**
Original speed	1	4	10 rpm	400	
Original Speed x .5	1	4	5 rpm	100	¼
Original speed x 2	1	4	20 rpm	1600	4

A change in RPM changes the force from imbalance by a factor of the RPM change factor squared.

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Types of Imbalance

Static Imbalance

Static imbalance is sometimes referred to as simple force imbalance.

A single heavy spot on the rotor is approximately equidistant from the bearings. If placed in frictionless bearings, the rotor would always stop with the imbalance mass at the lowest point. A correction mass can be placed at the opposite side of the rotor. The rotor can then be spun by hand to see if the correction was sufficient or if additional mass is needed to further reduce the imbalance.

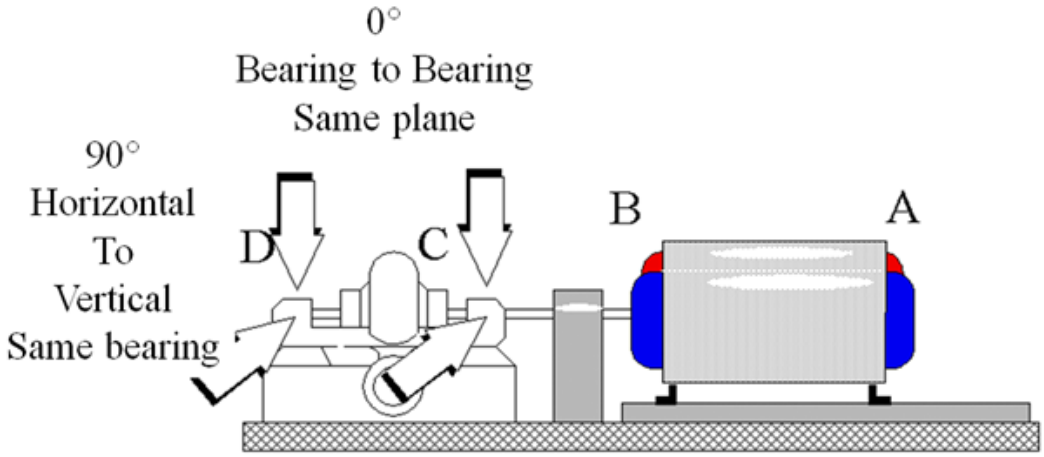
This method can be effective on narrow rotors, such as car and motorcycle tires and thin profile fans and pump impellers.

Static imbalance can be corrected with a single plane balance correction.

Characteristics of Static Imbalance

The 1 x TS amplitude tends to be highest in the radial direction on center-hung machines. A horizontal to vertical ratio should not be more than 3:1. Axial 1 x TS amplitude is generally low relative to the radial readings. When imbalance is the dominant forcing function, the 1 x TS amplitude approximates that of the overall value.

Vibration at 1 x TS peak can be generated from a host of sources. Occasionally, phase data can be very useful in verifying imbalance versus other forcing functions. The phase relationship on a statically imbalanced machine is usually uniform from end to end of the rotor; 0 degrees phase shift from inboard to outboard on the same plane. Expect a 90 degree phase shift from horizontal to vertical at the same bearing location.



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Dynamic Imbalance

Dynamic imbalance is virtually every other case of imbalance other than static imbalance. Couple imbalance, over-hung rotors and vertically mounted machines are also types of dynamic imbalance. Dynamic imbalance requires two or more correction weight planes to balance the machine.

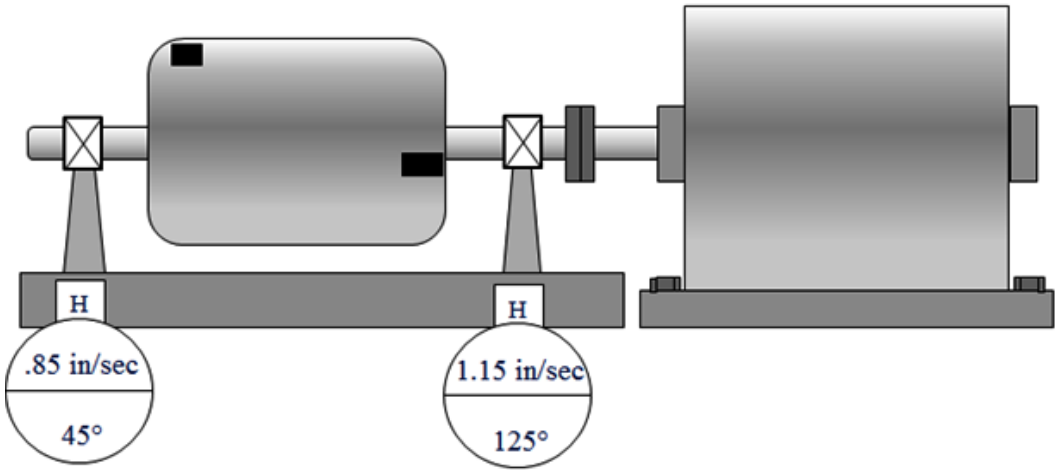
Characteristics of Dynamic Imbalance

Like static imbalance, the 1 x TS amplitude is usually highest in the radial direction on center-hung machines. A horizontal to vertical ratio of 3:1 or less is typical. Axial 1 x TS amplitude is generally low compared to the radial readings. When imbalance is the dominant forcing function, the 1 x TS amplitude is close to that of the overall value.

The phase relationship of a dynamically imbalanced machine is seldom uniform from end to end of the rotor. Almost any phase relationship from inboard to outboard on the same plane is possible.

The phase relationship from horizontal to vertical at the same bearing location should still be close to 90 degrees.

Given unequal mass(es) on the inboard versus the outboard locations, 1 x TS amplitude will generally be higher on the location that has the greater mass displacement. In the event that the mass(es) are not separated by 180 degrees, the phase data will reflect a similar relationship.



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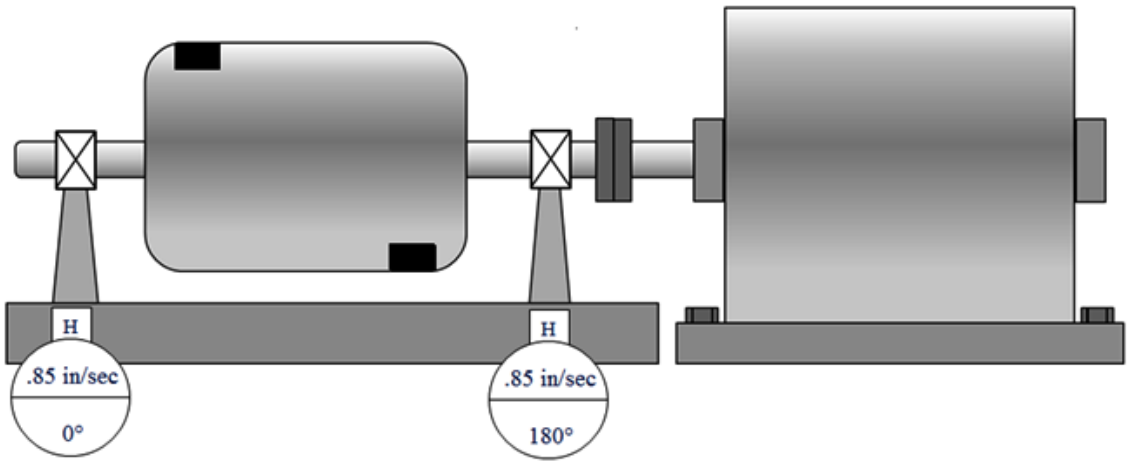
Couple Imbalance

Couple imbalance is still considered dynamic imbalance. However, in the relatively rare case of couple imbalance, there are two masses of equal value located exactly 180 degrees apart on opposite ends of the rotor.

Couple imbalance requires two or more correction weight planes to balance the machine.

Theoretically, the amplitude at 1 x TS will be equal on each bearing location.

The phase relationship from end to end should be 180 degrees.



Note

The dynamic properties of the structure (and the individual components) can have a significant effect on the amplitudes and phase readings.

Over-Hung Rotors

Over - hung rotors suffering from static imbalance can be corrected with a single plane balance job. Dynamically imbalanced over-hung rotors will require two or more correction weight planes.

The 1 x TS vibration will be elevated in the radial direction(s) similar to center-hung rotors. However, it is very common to see elevated 1 x TS energy in the axial direction as well. This is usually due to axial loading (thrusting) from the imbalance force(s).

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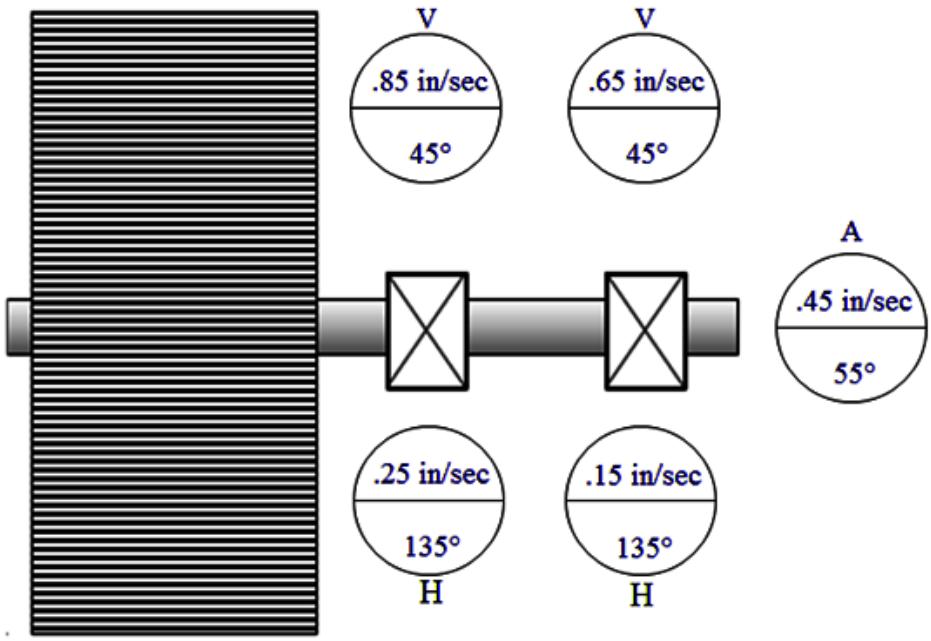
Thoroughly examine the phase data in all three axes to determine whether a single or dual plane correction is in order.

If the phase readings from end to end are the same on the same plane in each direction (vertical to vertical, or horizontal to horizontal), a single plane balance job will probably be sufficient.

Consider the axial phase reading on each bearing. If the 1 x TS vibration is due to imbalance, both bearings will be in phase.



Vertical Equipment

Vertical equipment can be challenging, since the imbalanced rotor (sump pump impeller, for example) is not accessible. Often, balance weights can be placed in a plane at the top of the motor, so that a centrifugal force creates a pull, counteracting the vibration at the top of the motor.

Sometimes, when the outboard (top) of the motor has an anti-reversing ratchet assembly, the bolts can be used as discrete weight locations. This is typical of hollow shaft pump motors.

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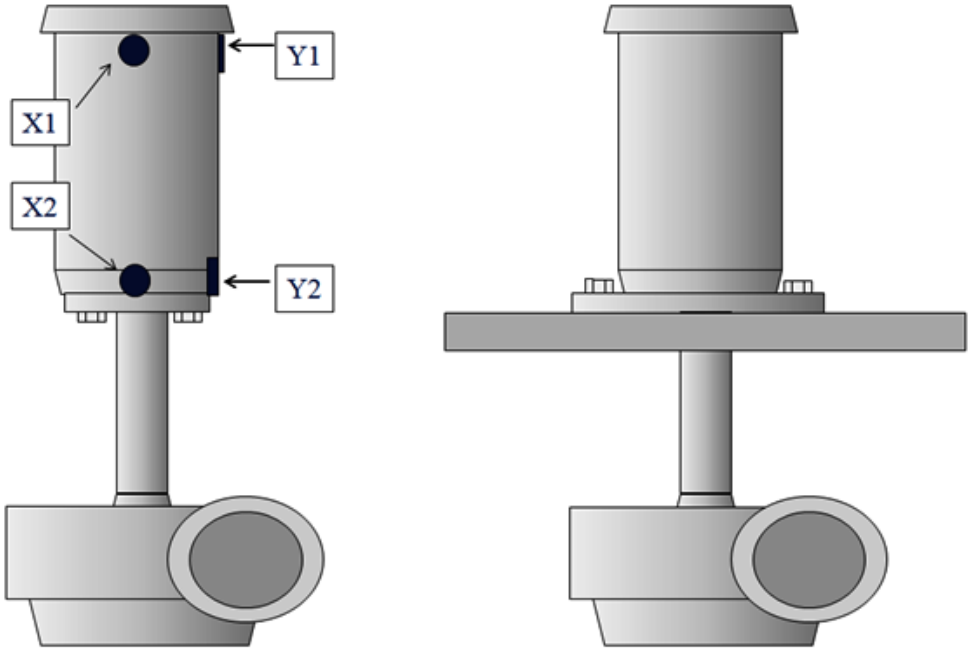
High turning speed vibration tends to be very directional on vertical machines. Compare the two radial (X, Y) direction amplitudes to eliminate the possibility of resonance prior to starting the balance job.

Vertical pumps tend to have significant stiffness differences, depending on the orientation of the inlet, or discharge port locations.



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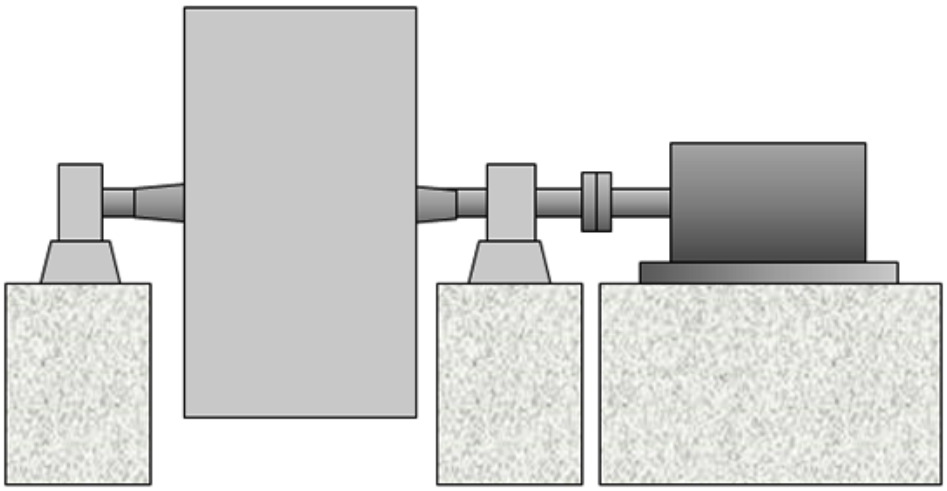
Case Histories – South #1 Kiln Fan

Machine	Horsepower	Speed	Fan Bearings	Fan Type	Mounting
South #1 Kiln Fan	350	1185	Fluid Film	Centrifugal - 8	Center - hung



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This fan is used on a kiln at a paper mill. Non-contacting displacement probes were not installed on the fan bearings. Vibration data was collected with accelerometers. Therefore the amplitudes are questionable. However, condition based monitoring is still a viable practice.

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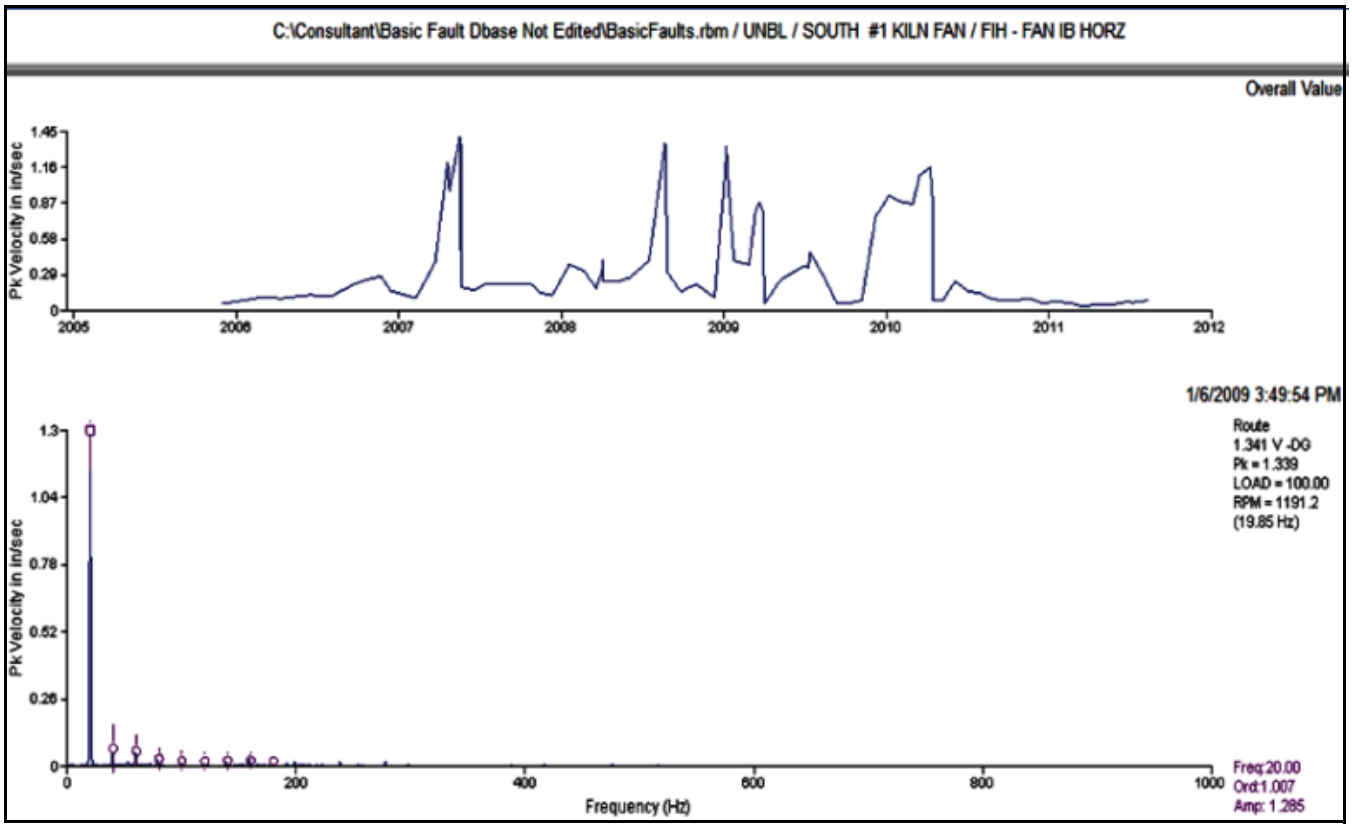

Point	Date	Trend	Amplitude of 1X
FIH	1/6/2009	Overall	1.31”/Sec

The dominant peak is turning speed. Any harmonics are extremely low in amplitude. Imbalance is suspected.

Always consider time domain data before reaching a conclusion.

In this case, the waveform is, indeed, periodic at rotational frequency. This is more evidence of an imbalanced condition.

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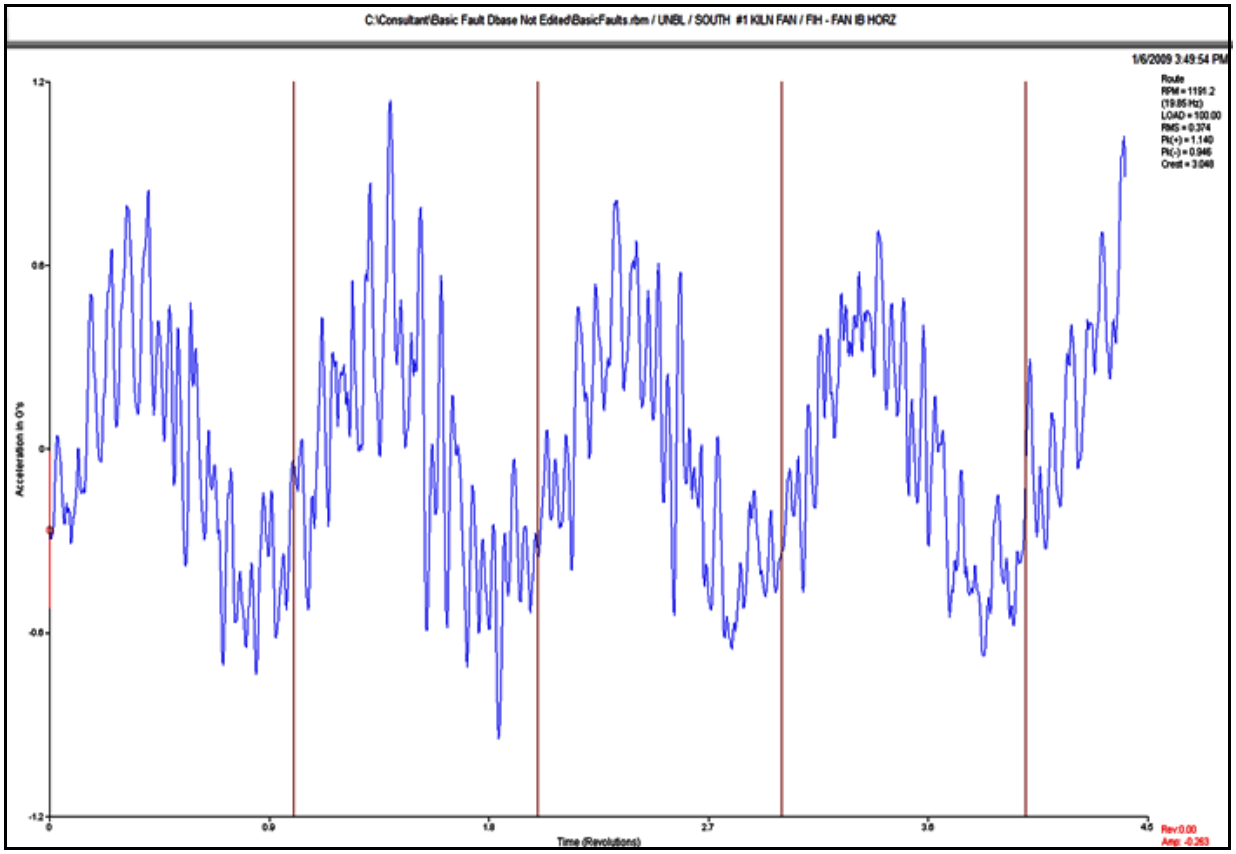
The waveform on the inboard horizontal point is periodic at rotational frequency, even in acceleration.



There is still evidence of some high frequency vibration in the system. This could be coming from the coupling or low level looseness. Sometimes vibration energy coming from the bearing of the driver can transfer to the driven unit.

Remember the waveform patterns to look for:

- Periodicity - repetitiveness
- Indication(s) of impacting
- G swing
- Evidence of high frequencies – multiple signals per revolution
- General shape – truncation, offset, etc.

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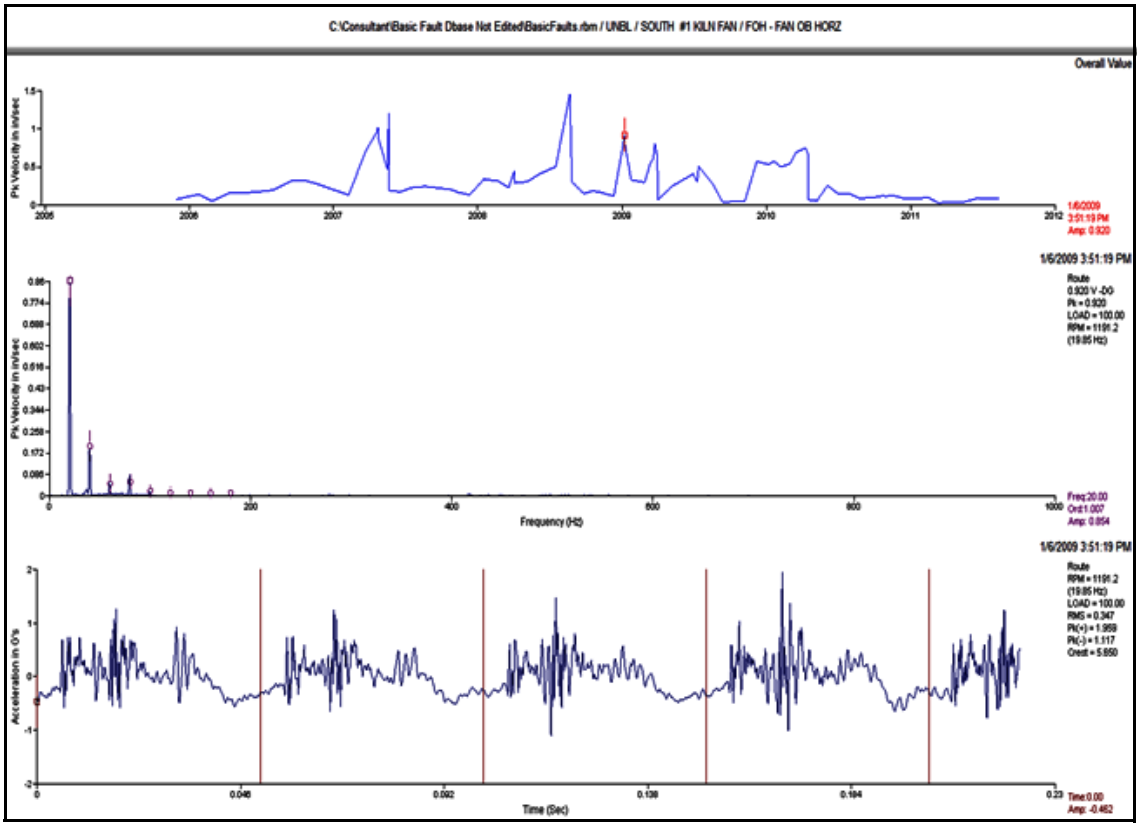
On both horizontal points, 1 x RPM is the dominant spectral peak. The outboard waveform is indicating some high frequency energy, possibly due to coupling wear or some other looseness.



Point	Date	Trend	Amplitude of 1X
FOH	1/06/2009	Overall	.85”/Sec

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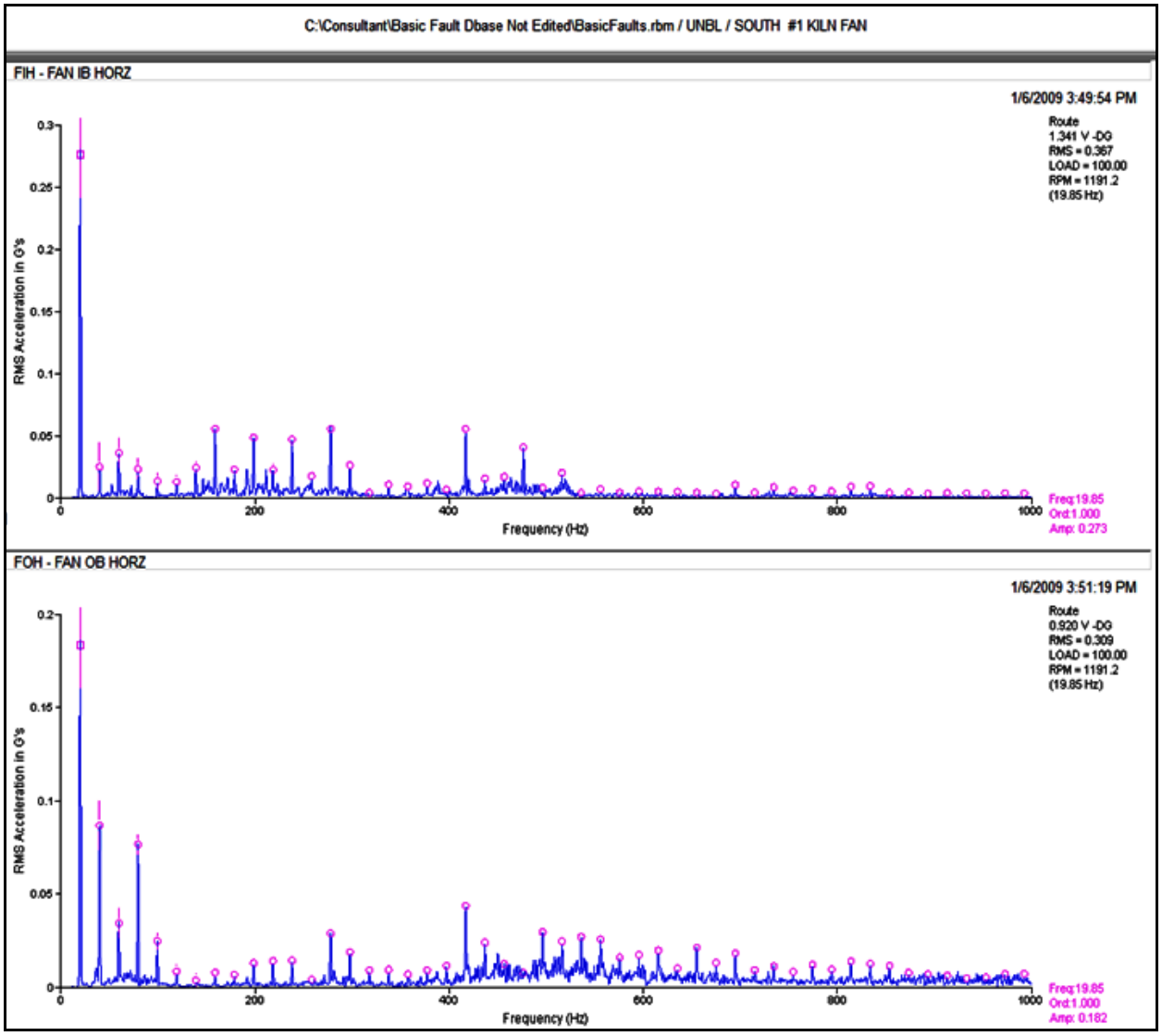
Viewing the spectra in acceleration helps to see these high frequency harmonics. Since this is a sleeve bearing machine, a certain degree of looseness is expected.



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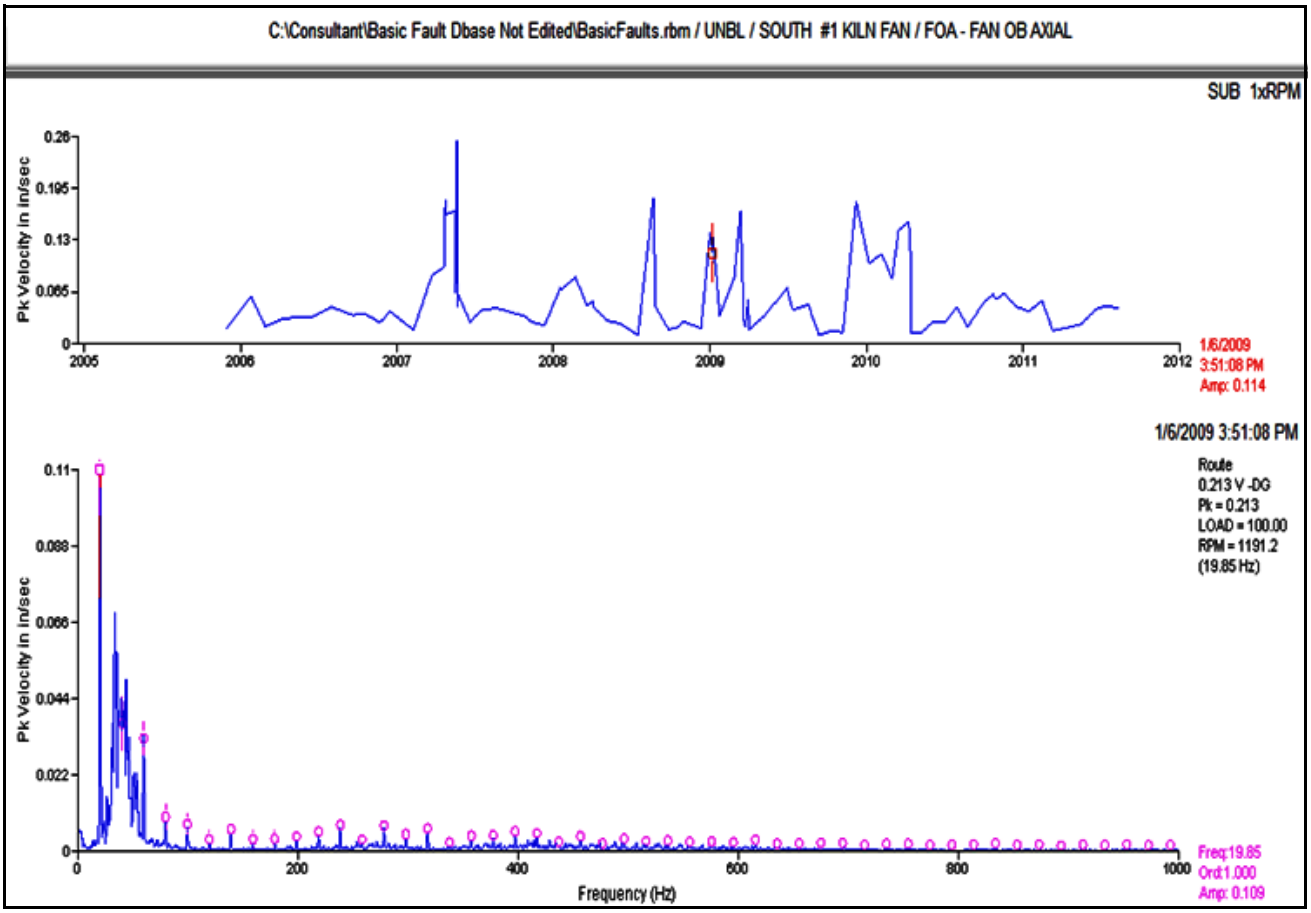
Common to most center-hung machines, little axial energy is present as indicated in the Fan Outboard Axial data point.



Expect to see more axial energy from imbalance on over-hung machines.

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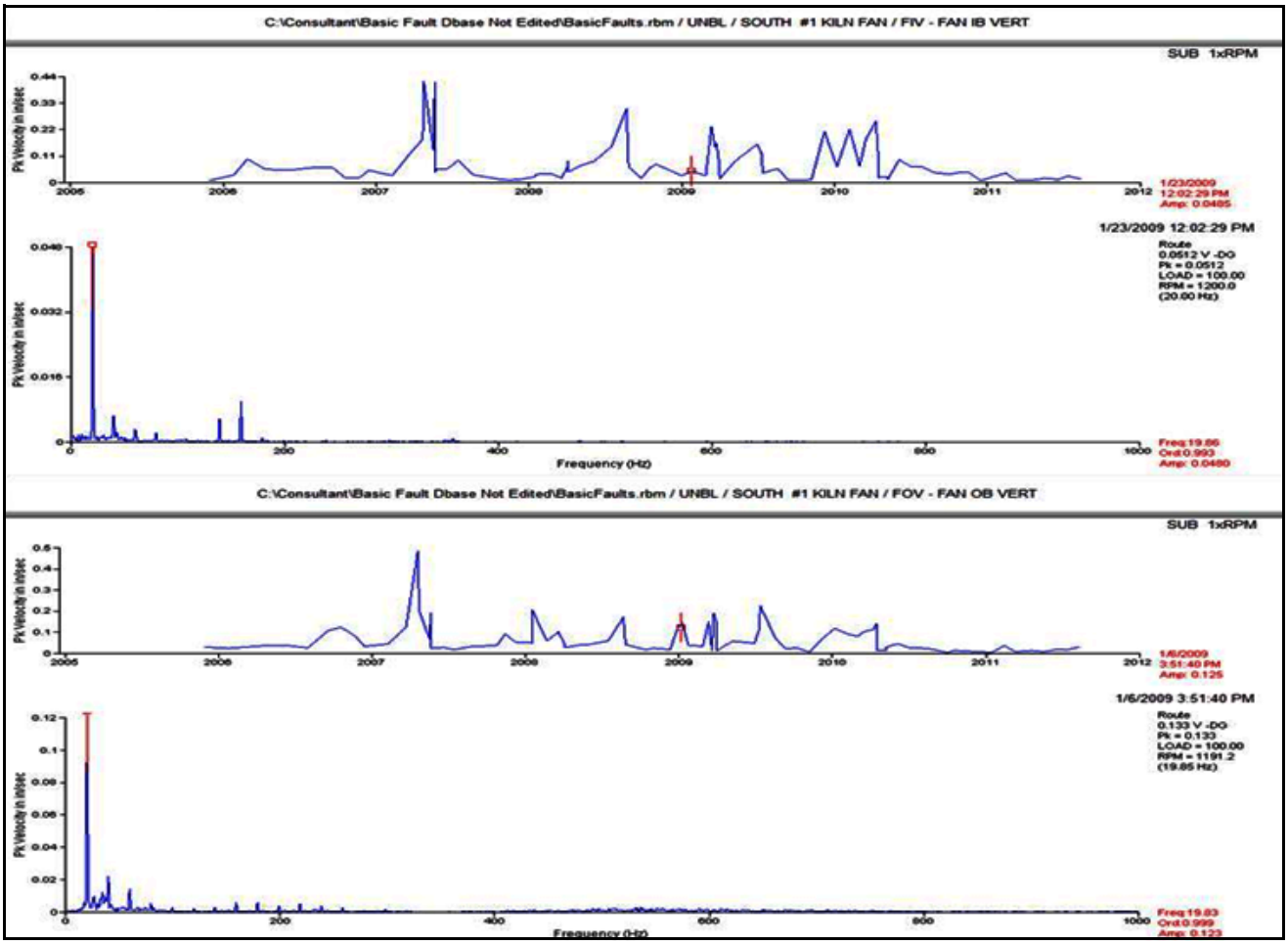
For unknown reasons, data was not collected on the FIV point on 1/06/2009. Therefore, January 23rd data is used in the example.

Data from the vertical fan points compared to the horizontal points indicate a greater freedom of movement in the horizontal direction. This is typical of an imbalanced fan.



Points	Date	Trend	Amplitude of 1X
FIV	1/23/2009	Sub – 1xRPM	.048”/Sec
FOV	1/06/2009	Sub – 1xRPM	.123”/Sec

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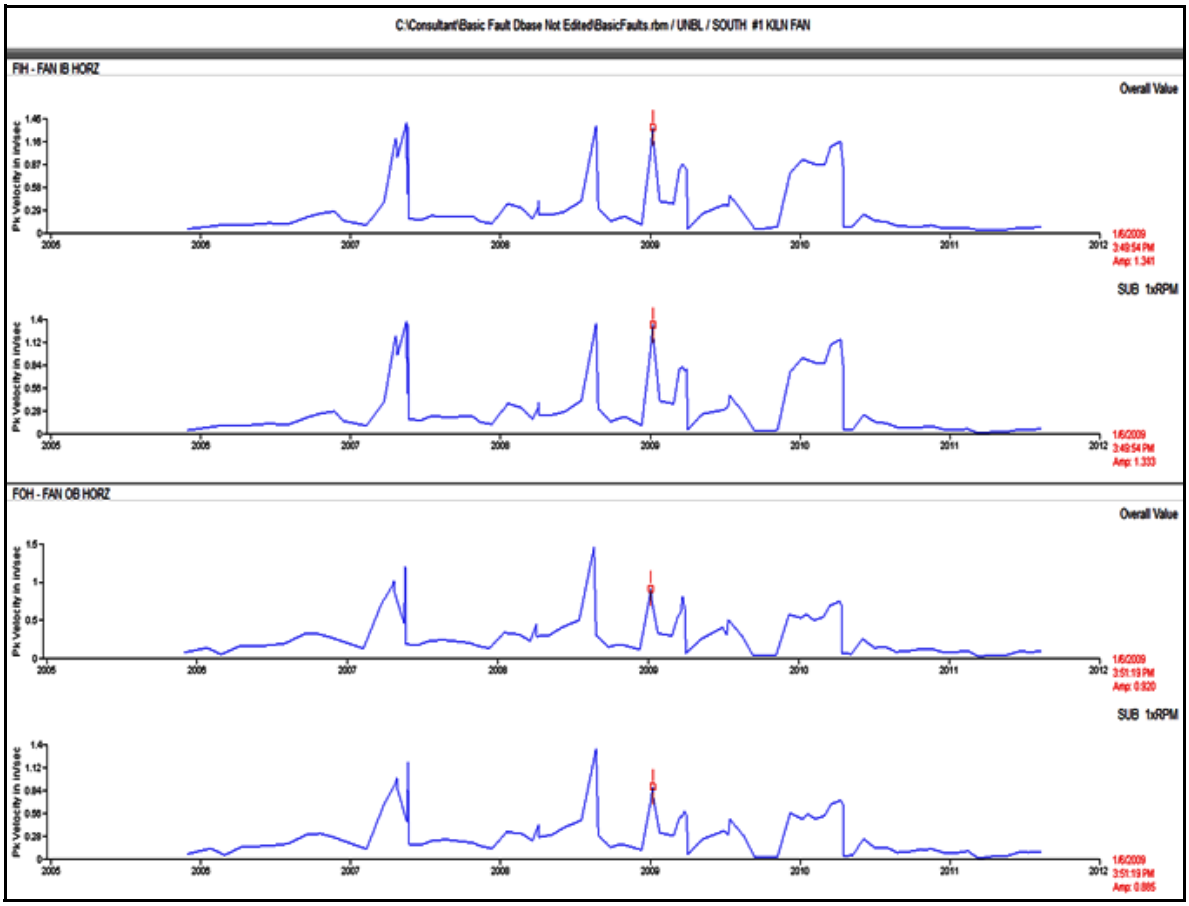
Trends of Overall and Sub – 1 x TS from the Fan Horizontal points indicate that most of the energy in the system is coming from 1 x TS. Fluctuations in the amplitude could be attributed to periodic cleaning or random release of material build up.



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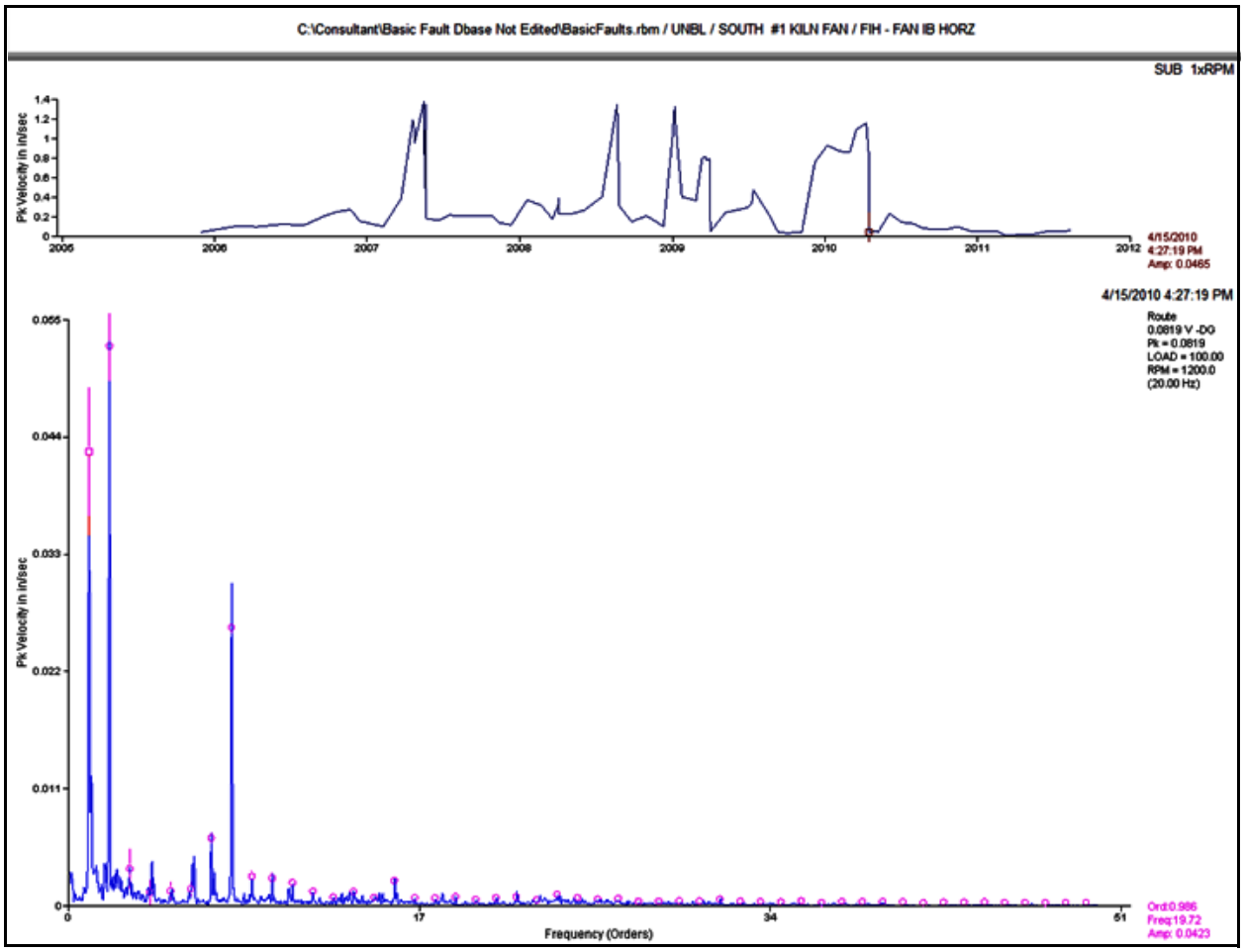
This unit was balanced in April, 2010. The amplitude of vibration at 1 x TS was reduced by approximately 97 percent, from 1.3 in/sec to .04 in/sec. It remained at acceptable levels for at least 16 months.



Some misalignment is indicated by the presence of the apparent 2 x TS peak. Blade pass frequency is showing up at 8 orders.

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Imbalance – Summary

Imbalance is usually the simplest forcing function to identify in vibration data, provided the RPM is known.

However, other defects (forcing functions) also cause vibration at that same frequency.

Collect phase data to confirm apparent imbalance from spectral and waveform data.

Examine the data carefully and collect additional data to verify any fault, especially when 1 x or Overall triggers an alarm.

If allowed to operate in an imbalanced condition for extensive periods of time, other machine components such as bearings, mounts, etc., may suffer damage.

Machine operators are a valuable source of information. Ask them:

- Have any operational parameters changed?
- What was the last thing done to the machine?
- Was it vibration related?
- Have you noticed unusual noise, or vibration?
- When was the last time any corrective action taken?
- When was it cleaned last?

Occasionally corrective actions have been performed on the machine and no one was notified.

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Review Questions

1. What is imbalance?

2. Imbalance is identified in the spectrum:

   A. At blade pass frequency.

   B. At 2 x TS.

   C. At 1 x TS.

   D. When harmonics of turning speed are present.

3. Which amplitude units are typically used when balancing in the field?

4. The rotational frequency of a fan is 1000 RPM. The amplitude of the turning speed peak at 1000 CPM is 2.6 mills caused by imbalance. The speed is then increased to 2000 RPM. The amplitude of the turning speed peak at 2000 CPM would be approximately _____ mils.

5. Although an imbalance condition produces primarily radial forces at 1 x RPM, an increased 1X RPM vibration may be present in the axial direction when the rotor design is _____.

6. True or False: When collecting Peak and Phase data to correct an imbalance condition, the vibratory high spot always represents the location of the heavy spot.

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7. The lag angle of a machine running above its first critical speed will be between _____ and _____.

8. The lag angle of a machine running below critical speed will be between _____ and _____.

9. The lag angle of a machine running at critical speed will be very close to _____.

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Single Plane Balance Procedure

Section 5

Objectives

- Preparation for the balance job.
- Explain the procedure.
- Define system properties:
  - Influence coefficient
  - Heavy spot
  - System lag
  - Vibratory high spot
- Demonstrate a single plane polar plot balance job.

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Single Plane Balance Procedure

Before performing any corrective action, identifying the problem is imperative. Careful analysis of all the available data should result in a confident diagnosis of the problem.

Conduct a visual inspection of the machine. Identify loose fasteners, housings, missing shims, cracked grout, broken welds, etc.

Balancing is difficult if the machine is not structurally sound. Correct looseness and other serious conditions, if at all possible. While balancing may reduce the effects of looseness, it does not correct the problem.

If the looseness is severe enough, balancing may be next to impossible due to the erratic nature of the phase relationship. If phase is erratic, the ability to accurately measure amplitude and phase is compromised.

Data Analysis

Collect high resolution data to help substantiate the problem. Occasionally misdiagnosis occurs because of poor resolution. Belt defects, transferred vibration and energy from rotor bar defects may contribute to the amplitude of an apparent turning speed peak.

Cross channel phase is very useful, especially when shutting the machine down is inconvenient. Phase data will often help resolve imbalance versus misalignment.

In the event of multiple defects on the machine, it is usually best to address the worst condition first. If turning speed harmonics are present in the spectrum, suspect looseness. If there are sidebands close to 1 x, suspect rotor bar problems in induction motors.

When imbalance is the dominant forcing function, there will be little to no turning speed harmonics.

Determine the ratio of the 1 x peak amplitude horizontal to vertical. If that ratio is 3:1or higher, suspect resonance. Collect as much data as possible to rule out resonance as a condition. If the machine is running close to or at resonance, balancing the machine will be extremely difficult.

Overhung machines will usually exhibit a 1 x component in the axial direction.

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The time waveform should be sinusoidal in nature with one major event per revolution. It should not be truncated or discontinuous. A beat in the waveform indicates the presence of a closely spaced frequency. Beat frequencies can make balancing very difficult.

The phase readings should be approximately 90 degrees horizontal to vertical on the same bearing and should be steady within 15 degrees to 20 degrees. Unsteady phase readings could indicate looseness or resonance.

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Procedure

After verifying the need to balance the machine, it is important to consider everything associated with the balance job.

- SAFETY
- Lock out / tag out policy
- Contact with the control room
- Limited starts on large horsepower motors
- Environmental impact on humans
- Impact on other equipment
- Assistants – you will need help
- Deadlines – this adds pressure
- Emergency contingencies
- Record keeping and Reporting
- Post balance data collection
- Archival of the entire job

Make a list of the items needed for the job.

Personal Protective Equipment	Protractor
Trial weights	Clip board
Correction weights	Calculator
Fasteners	Data collector
Scales	Sensors and cables
Polar plot paper – several sheets	Tachometer / cable
Colored pencils	Reflective tape
Note pad	Adapters
Parallels	12” scale
Radio / cell phone	Pencil sharpener

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Challenges Involved in Balancing:

All corrective actions have potential for errors. Understanding the sources of those inherent to balancing can improve the ultimate balance solution.

1. Estimating the light spot on the rotor.

- A common mistake is forgetting to add or subtract 180 degrees after estimating the heavy spot.
- Balancing in units other than displacement involves a unit's phase difference.

2. Plotting the vectors.

Note

A **vector** is a quantity having direction as well as magnitude, especially as determining the position of one point in space relative to another.

- Proximity – solutions are difficult to determine when two vectors are too close together.
- Length – i.e. if the Trial Run vector is too short, the RT vector is difficult to draw.

  Draw an arrow at the end of every vector.

  Label every vector.
- The direction of **RTT** is **critical**.

3. Correction weight calculations.

4. Determining the correction weight location.

- Collapse the **RTT** vector towards the **back** of **R**.
- The angle between these determines the amount of movement for the correction weight.

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5. Amplification factor.

- Proximity of TS to resonance affects the amplitude of measured vibration.
- As TS approaches resonance the amplification factor increases.
- As TS increases above resonance the amplification decreases.

Note

A detailed explanation of each vector type will be included in the example balance job.

6. Determining system lag.

- Always be between 0 degrees and 180 degrees.
- Estimation:

  If TS is below resonance, estimate at 45 degrees.

  If TS is above resonance, estimate at 135 degrees.
- System lag angle depends on amount of damping.

  Calculating system lag helps prevent adding another trial weight.

  Determines the location of trim weights.

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Preparation

Thoroughly clean a dirty rotor. Occasionally, cleaning is all that is required for bringing the machine into acceptable operating levels. After cleaning, inspect the rotor for obvious structural problems such as cracks, worn surfaces, etc.

While the machine is not in operation, measure the radius of the weight location. Determine the best and safest method of attaching trial and corrective weights. Try to determine the weight of the rotor.

Securely attach transducers to each bearing. Attach the tachometer so the leading edge of the reflective tape will trigger the once-per-revolution pulse in line with the first sensor.

If at all possible, ensure that the rotor will be balanced at operating frequency. Variable frequency drives create a unique challenge to the balance technician.

Locate all personnel, tools, parts, etc. in a safe location before beginning the balance job.

Balancing Overview:

Collect reference peak/ phase data.

Stop the machine.

Attach trial weight on the estimated light spot.

Start the machine – collect trial run peak/ phase data.

Determine correction weight and location.

Attach correction weight.

Balance is complete in the amplitude is in tolerance.

Perform a trim run, if necessary.

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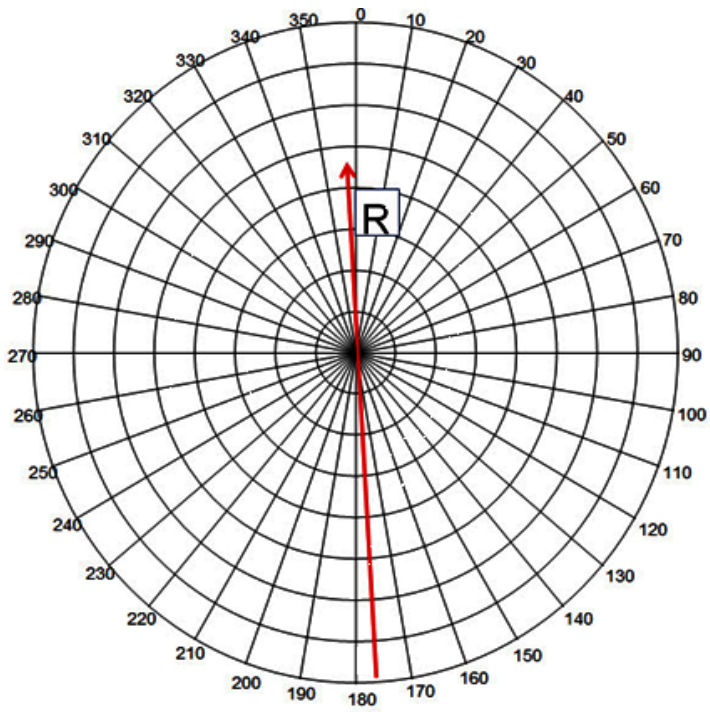
Reference Run

The reference run is the **as is** condition. The amplitude of the reference run will determine the value of each division on the polar paper. For example, 1 division = 1 mil(s). Plot the **R** (reference) vector on the polar paper. Extend the back of **R** to the outer edge of the outer most division on the paper. **Label each vector**. In graphical balancing, a vector represents vibration amplitude and phase. In this example, the reference amplitude was 4.5 mils at 357 degrees.



Attach the trial weight. Estimate the trial weight amount.

Estimated trial weight ≈ 56,333 x (rotor weight /ω^2) oz.-inches.

The location can be estimated using the equation:

Heavy spot ≈ Angle between tach and 1st sensor + θ (phase reading) + units adjustment – lag angle*.

*If 1x is below critical, use 45 degrees.

*If 1x is above critical, use 135 degrees.

Caution!

The estimated **light** spot will be 180 degrees opposite the estimated **heavy** spot.

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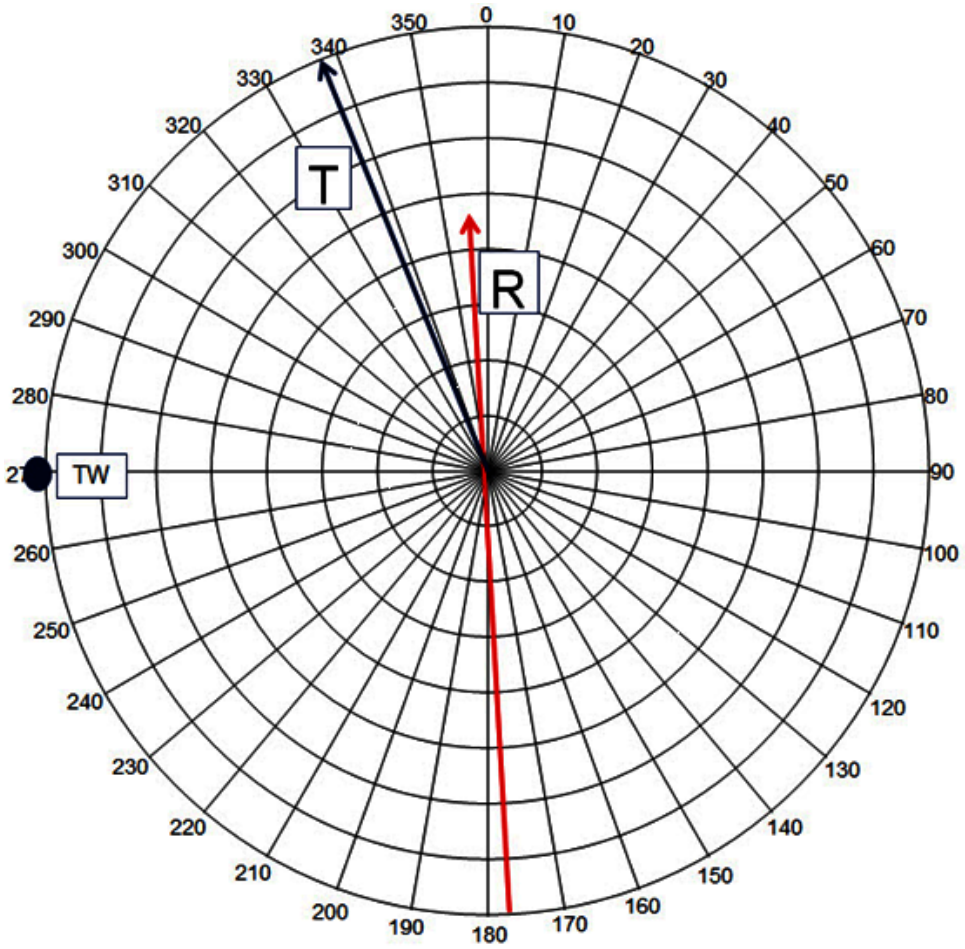
In the example, 1 ounce was attached at 270 degrees.

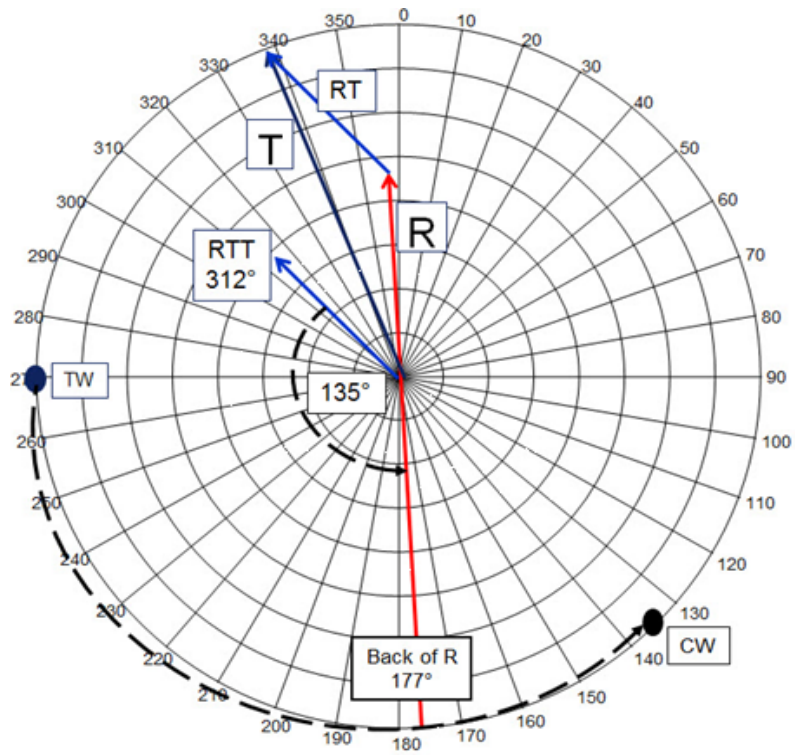
Record and plot the T (trial) run vector.

In this example, the trial run generated a vector at 337 degrees with amplitude of 8 mils.



Connect **R** to **T** from the tip of **R** to the tip of **T**. This is the **R - T** vector.

Transpose the **R - T** vector to the origin or center of the polar paper. This is the **R - T_T** vector. **R - T_T** must be the same length of and parallel to **R - T**.



The phase angle of the R - T_T vector is measured as 312 degrees. The amplitude is approximately 3.75 mils.

Use a protractor to measure the angle between RT_T vector and the back of the R vector. Subtracting the phase reading difference between RT_T and the back of R will produce the same angle. In this example, 312 degrees - 177 degrees= 135 degrees. This is how far in degrees the correction weight location is from the trial weight location. The direction is the same direction that the RT_T vector is collapsed to the back of R.

The correction weight in this example needs to be placed 135 degrees CCW from the trial weight location, which was 270 degrees. The correction location is at 135 degrees. (270 degrees – 135 degrees = 135 degrees)

The correction amount is calculated using this formula:

CW = TW (R/RT)

Where TW = 1 ounce, R = 4.5 mils and RT = 3.75 mils.

1 x 4.5 / 3.75 = 1.2

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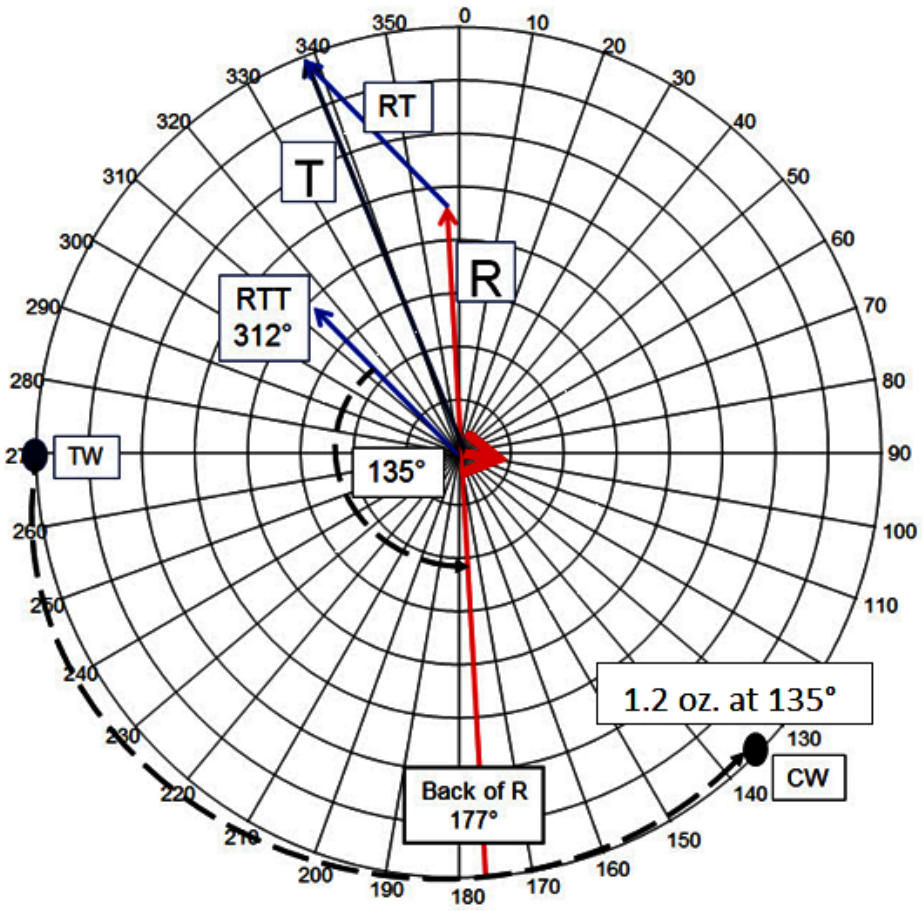
Attach 1.2 ounces at 135 degrees.

The solution was correct. The correction run generated a peak / phase reading of .77 mils at 100 degrees. No further runs are necessary.



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Summary

High resolution data is very important. Other mechanical forcing functions can generate high amplitudes at 1 x RPM. Often, a harmonic of a sub-synchronous peak appears to be a high amplitude peak at 1 x RPM.

Scrutinize the vibration data. Consider direction, ratio, etc.

Collect additional data such as phase readings. Talk to the operator. Find out if any operating parameters have changed. Remember, the only operational parameter that can change the force of imbalance is the rotor RPM.

Peak and Phase readings identify the vibratory high spot, not the rotor heavy spot.

Conduct structural inspections. Identify weaknesses, cracks, missing bolts, missing shims, etc.

Balance the machine only if the primary cause of excessive vibration is imbalance.

Do not get in a hurry. “Measure twice and cut once”.

Safety is the primary concern.

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Review Questions

1. When using a polar plot for a balance procedure, what two values does a vector represent?

2. True or False: When correcting an imbalance condition, looseness and/ or other vibration related problems should be corrected after the balance procedure.

3. The most important issue to be concerned with when performing a single plane balance job is _____?

4. The turning speed of the machine is 1740 RPM. The bump test results indicate a natural frequency at 13 Hz. What value should be used to estimate system lag?

5. The turning speed of the machine is 1740 RPM. The bump test results indicate a natural frequency at 41 Hz. What value should be used to estimate system lag?

6. When collecting phase data to confirm an imbalance condition, the phase difference between horizontal and vertical data should be _____.

7. During a single point single plane balance, the reference amplitude at 1 x turning speed was four mils. A trial weight of 11 grams was added to the rotor and a trial run was measured. After graphing the trial run, the line drawn from the reference vector to the trial run vector is six mils. How much correction weight needs to be added to the rotor after the trial weight is removed?

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Misalignment, Bent Shaft and Soft Foot

Section 6

Objectives

- Describe various types of shaft misalignment.
- Discuss the effects of misalignment on rotating equipment.
- Explain spectral and waveform patterns suggesting misalignment, bent shaft and soft foot.
- Determine corrective actions.

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Descriptions and Observations

Misalignment is the condition between two or more coupled shafts whose axes either intersect at an angle or are not perfectly parallel.

There are two types of shaft misalignment; angular and offset. Most coupled shafts operate under a combination of both – horizontally and vertically.

Ideally, two perfectly aligned inline shafts share a common axis. Belt drive shafts would be perfectly parallel. Input, intermediate and output shafts in many gearboxes would be parallel.

Any two coupled shafts are prone to misalignment, regardless of the coupling mechanism. Inline couplings, belt drives, fluid drives, etc. are all subject to a degree of misalignment. Machines with “C” face endbells are also susceptible to misalignment.

Perfect shaft alignment is impossible. Coupled shafts operate under a combination of some offset and angular misalignment. Therefore some level of compromise must be met with respect to nonlinearity between the coupled shafts. These specifications are usually in mils (thousandths of an inch), or millimeters of offset or angularity.

A good understanding of the thermal activity of each component in the drive train is vital to attaining an acceptable level of alignment. Frequently, the shafts in question must be misaligned significantly while at rest. While approaching normal operating conditions, the machine undergoes a certain amount of thermal activity, allowing the shafts to grow into alignment.

Knowing the type of coupling in the machine train can be very useful when diagnosing misalignment. However, some of these make misalignment diagnosis challenging.

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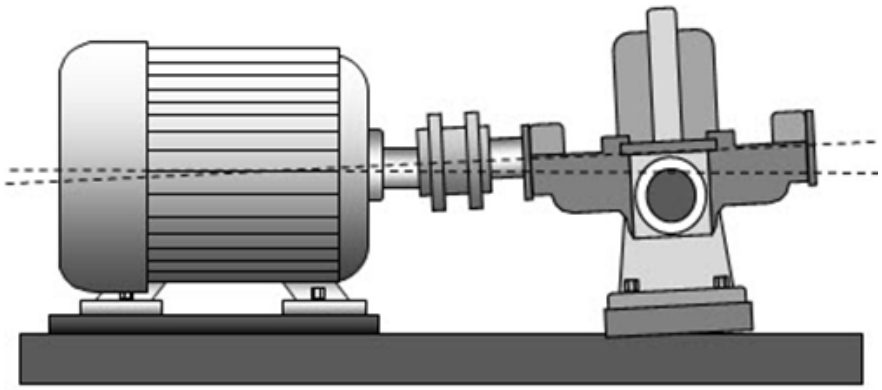
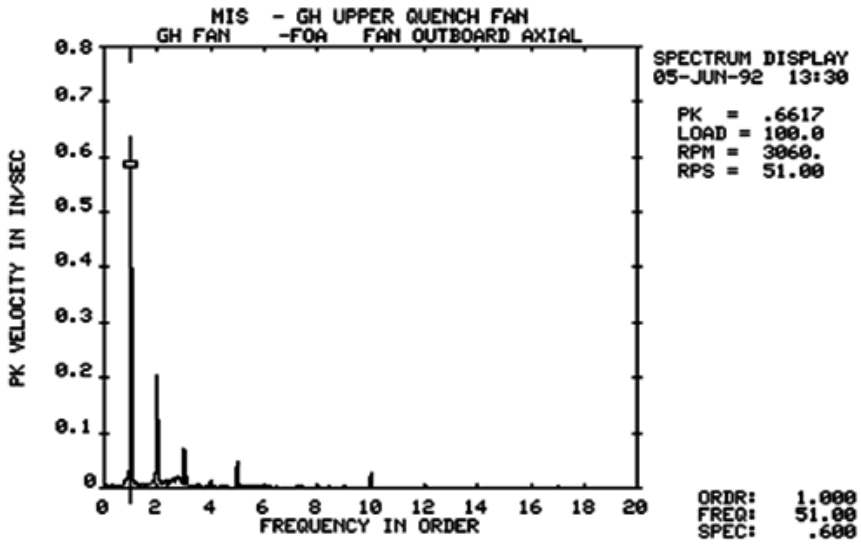
Angular Misalignment

Angular misalignment is present when the axes of two coupled shafts meet at an angle.

This angle may be in the vertical and/ or horizontal planes.



Angular misalignment causes excessive vibration amplitude at 1 x TS in both the radial and axial directions. A general rule of thumb often associated with angular misalignment states that if the vibration amplitude at 1 x TS in the axial direction is 50 percent of or higher than the amplitude at 1 x TS measured in the radial direction, a significant amount of angular misalignment exists in the system.



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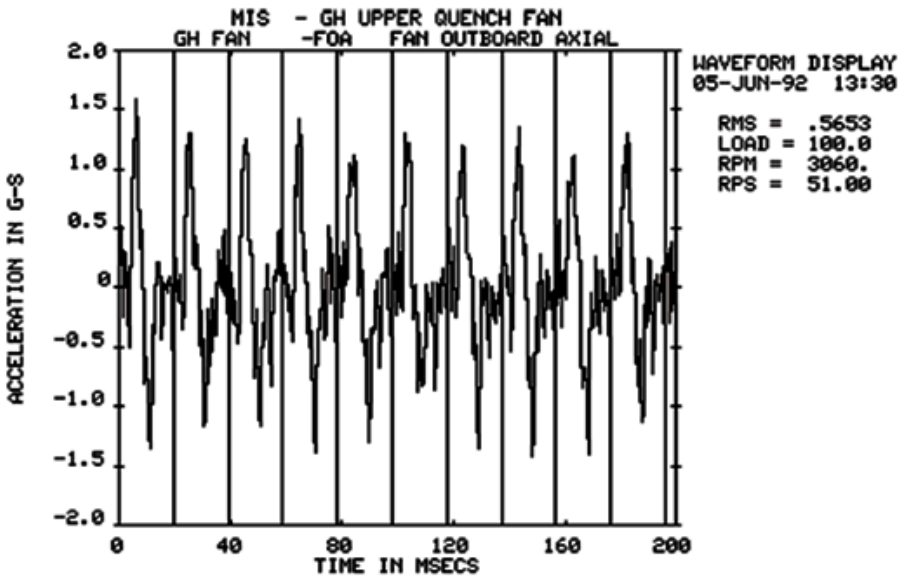
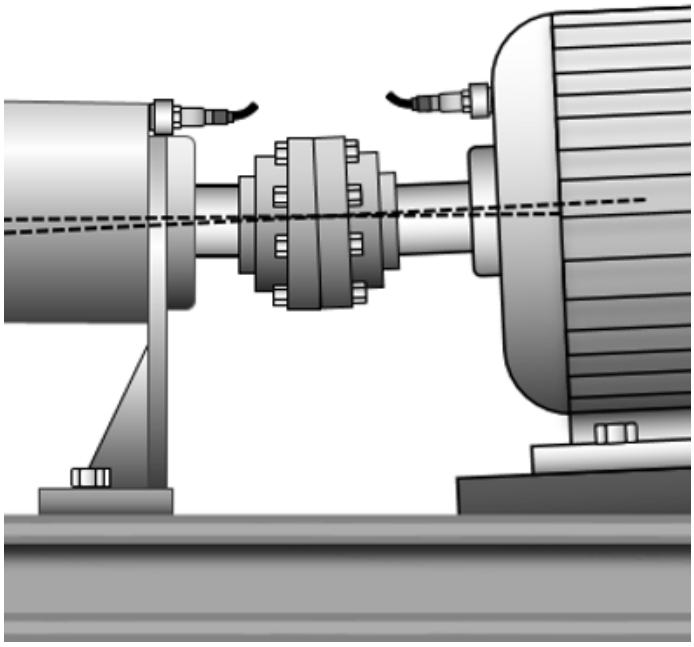
There should be one, two or more movements per revolution in the time domain with a repeatable, periodic pattern.



The highest vibration amplitude is usually in the direction of misalignment. If the angularity is in the horizontal direction, the highest amplitude is normally in the horizontal plane.

A 180 degree phase shift across the coupling will often help verify misalignment. With angular misalignment, this shift will be measured in the axial direction.



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Note

Remember to account for sensor orientation when collecting axial phase. Measuring phase from the inboard end of both units means the sensor is being rotated 180 degrees.

Misaligned shafts can cause subsequent damage to rotating equipment.

The inside of the coupling guard may be covered with grease.

Bearings can overheat, resulting in premature failure.

Frequent coupling failures have been documented.

Seals on either or both units may leak.

Extreme cases of misalignment have been responsible for shafts breaking; usually close to the inboard bearings.

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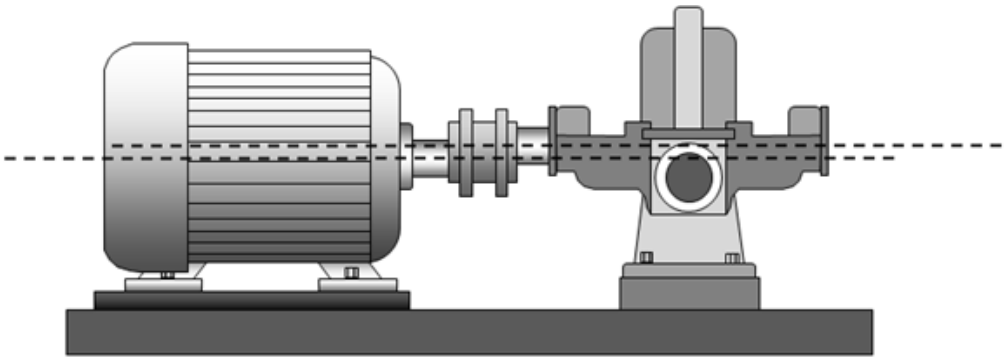
Offset Misalignment

In the case of offset misalignment, the coupled shafts are parallel but displaced from one another. This displacement can be vertical, horizontal or both.



A high amplitude of vibration at 2 x TS is one indication of offset misalignment. A general rule of thumb often associated with offset misalignment states that if the vibration amplitude at 2 x TS is 50 percent of or higher than the amplitude at 1 x TS, a significant amount of offset misalignment exists in the system.

The time waveform will show movement twice per revolution in the shaft.

When offset misalignment is present, force is transferred across the coupling offset, generating a movement twice per revolution of the shaft. 2 x TS amplitudes will usually be highest perpendicular to the offset. For example, if the shafts are offset horizontally, the highest vibration will usually be vertical, and vice versa.

With extreme offset misalignment, there is enough force generated to bend the shaft, leading to an elevated axial 1 x TS spectral peak.

Similar to angular misalignment, there will usually be a 180 degree phase shift across the coupling. When offset misalignment is dominant, this phase shift will show up when measured in the radial direction instead of in the axial direction.

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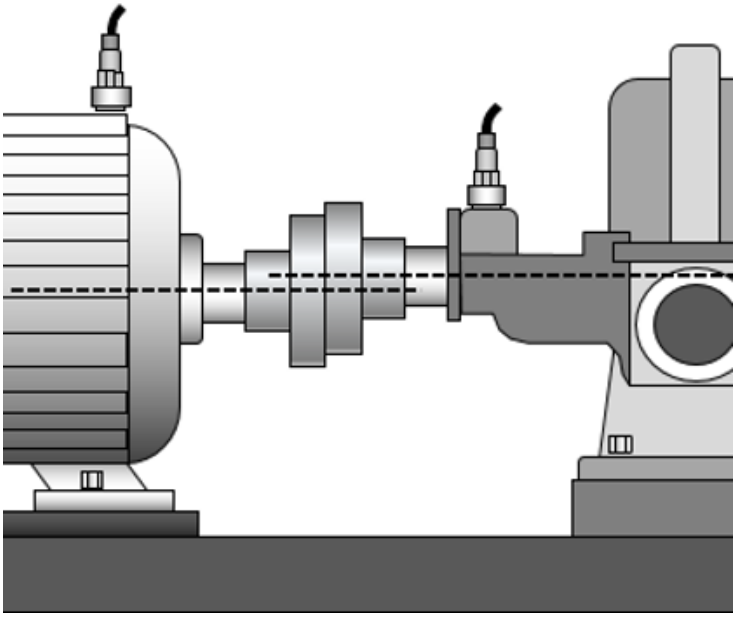

A coupling with multiple segments per half tends to generate higher amplitudes at turning speed harmonics. For example, a misaligned 3-Jaw coupling will result in an elevated 3 x TS spectral peak. The amplitude will depend on the severity of the misalignment and the condition of the rubber insert. Flexible – grid or flexible – gear type couplings may also generate greater amplitudes at a frequency of the number of segments times turning speed.

If coupling misalignment is the primary forcing function, the time waveform will be periodic and repeatable. The period for each event will be that of the number of segments times turning speed.

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Bearing Misalignment

There are several variations of bearings being misaligned. On a line shaft, one or more of the pillow block bearings could be set at an angle to the other bearings. Thermal distortions and soft foot could contribute to bearing misalignment. A motor bearing may be mounted improperly (cocked) on the shaft or in the housing. Excessive misalignment at a drive could cause the shaft to bend, inducing a cocked bearing.

Bearing misalignment should be considered any time there is a high amplitude of vibration in the axial direction at 1 x or 2 x TS. A peak in the spectrum at a frequency equal to the number of rolling elements in the bearing times the turning speed is also possible.

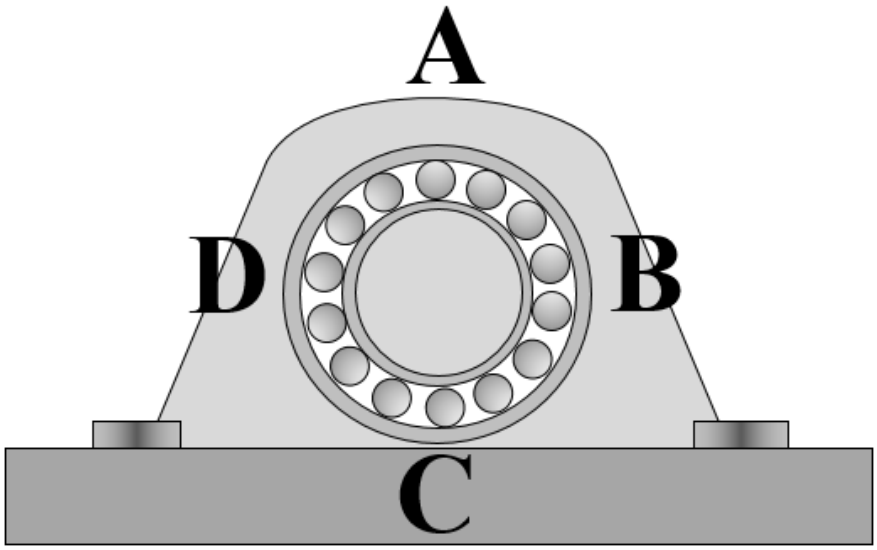
The time waveform often shows evidence of truncation or flattened pattern, indicating rubbing.

Collect phase data in the axial direction at multiple positions on the face of the bearing to help identify bearing misalignment.

Consider the face of the unit as a clock. Compare the phase readings from 12:00 to 6:00. Also compare the readings between 3:00 and 9:00.

In the following illustration these positions are A, B, C and D.

The phase relationship between A and C will be 180 degrees, and the relationship between B and D will also be 180 degrees (± 20 degrees). If the phase values from all four positions are within 20 degrees, the axial vibration is likely not caused by a cocked bearing or angular misalignment.



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Bent Shaft – Characteristics and Detection

The vibration data may resemble that of imbalance, misalignment or a combination of both.

A bent shaft actually causes imbalance. The direction of response can vary depending on the location of the bend. If the bend is near the center, expect a dominant 1 x TS peak radially and axially. A high amplitude at 2 x TS is possible when the bend occurs closer to the end of the shaft.

On a flexible rotor, as the turning speed begins to exceed first critical, the amplitude will increase exponentially. Decreasing rotational frequency will tend to cause the amplitude to decrease rapidly. If imbalance were the major forcing function, the amplitude would decrease gradually.

Phase data may be very useful in identifying a bent shaft. When the bend is near the center of the shaft, the phase readings will be 180 degrees out of phase axially across the bend, not the coupling. If the transducer is inverted 180 degrees to collect phase data from end to end, the actual reading will be in phase, not 180 degrees out.

Phase data collected on the face of the machine will resemble a cocked bearing. A severely cocked bearing can produce a bent shaft. The phase readings will be 180 degrees out top to bottom, and side to side. (12:00 to 6:00 and 3:00 to 9:00) If the end of the shaft is bent between the bearing and the coupling, expect the phase readings to be 180 degrees out of phase between the two inboard bearings.

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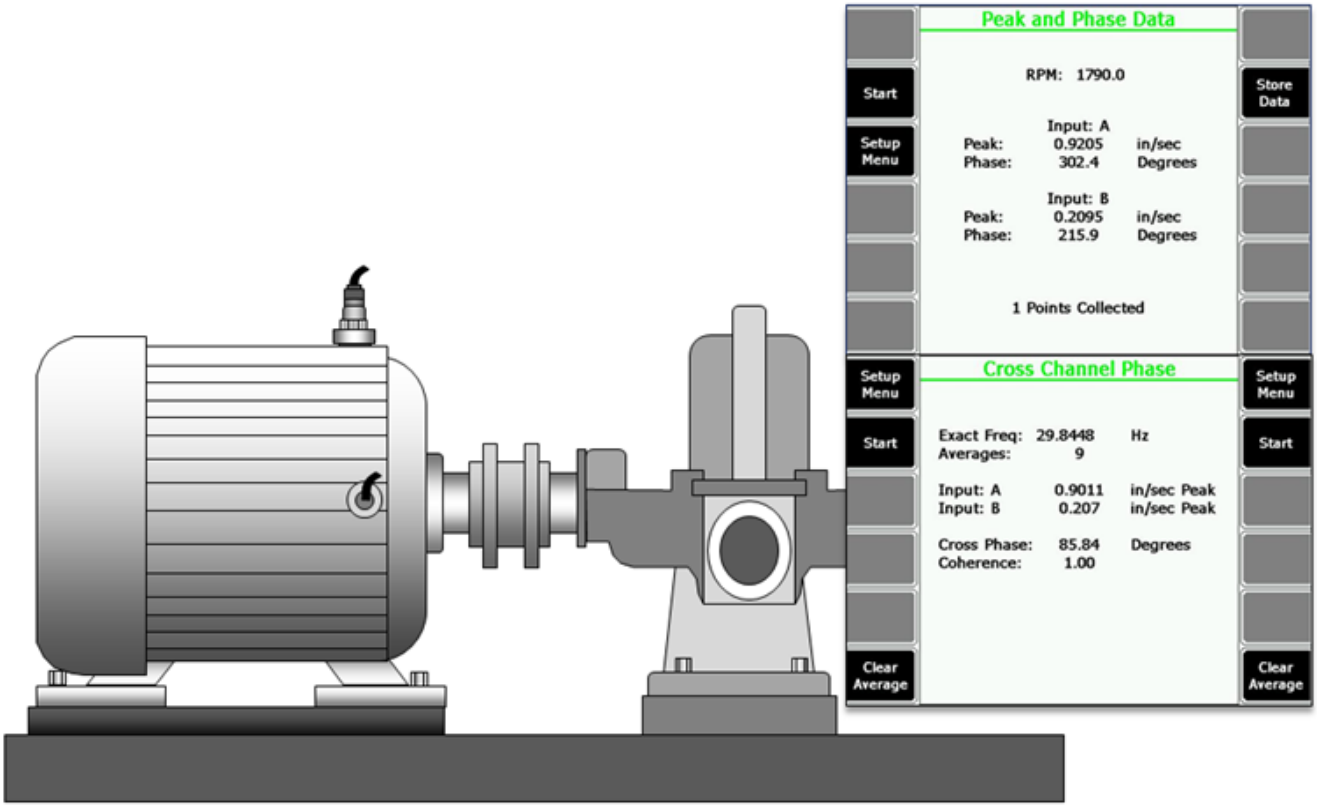
Imbalance will always generate a peak at 1 x TS. Without any additional forcing functions, the phase readings will be 90 degrees between the horizontal and vertical directions on the same end of the component.



A bent shaft will tend to mimic an imbalance condition. The bend in the shaft actually causes the rotor to be imbalanced. Balancing a rotor with a bent shaft can bring the vibration response down to acceptable levels. However, the shaft should ultimately be replaced to ensure safe and reliable machinery operation.

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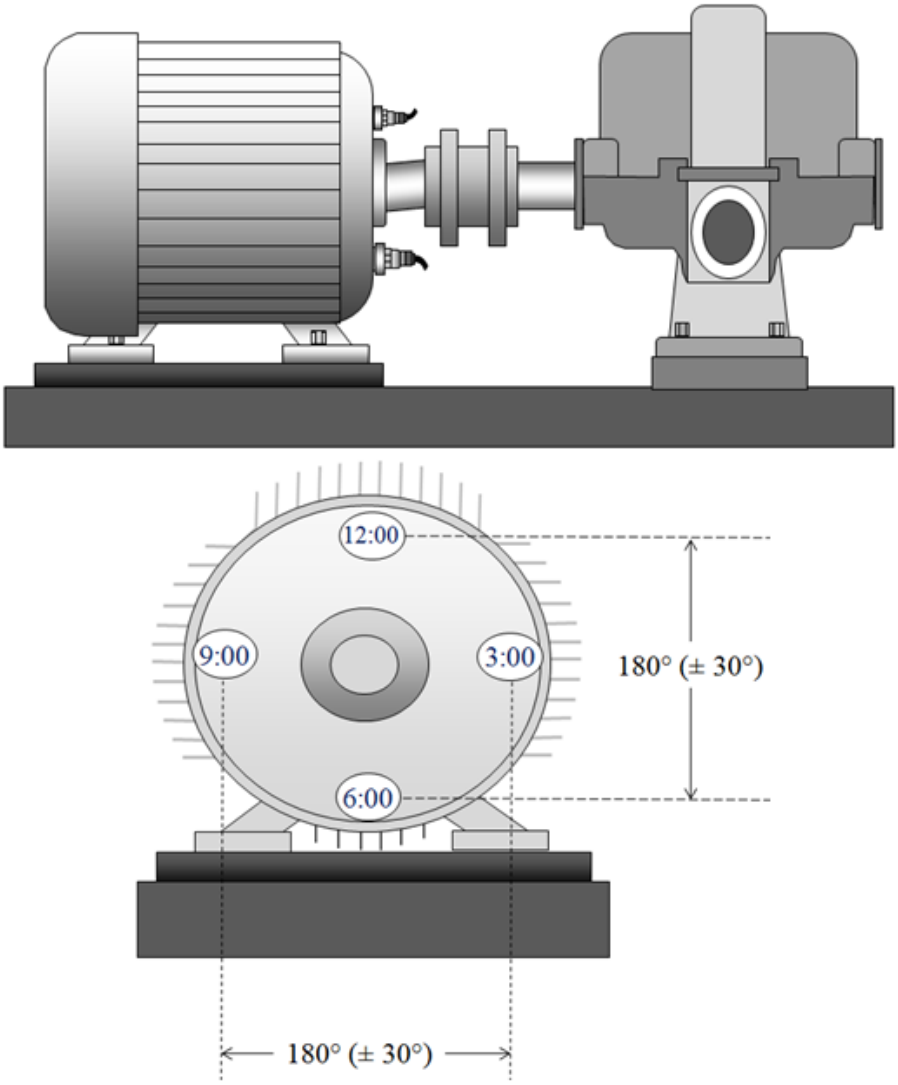
Bent Shaft – Phase

If the shaft is bent past the bearing, i.e. the coupling fit, expect a phase shift across the face of the bearing housing, or end bell. This condition may be difficult to prove. With the machine coupled, the data may resemble that of misalignment or imbalance. Properly aligning a machine with a bent shaft is very difficult, if not impossible. Under safe conditions, check the run-out of the suspect shaft with a dial indicator.



Measuring phase across the coupling may also be useful in identifying a bent shaft past the bearing. This can be difficult with guards in place. The phase readings tend to be 180 degrees between the inboard horizontal or inboard vertical positions.

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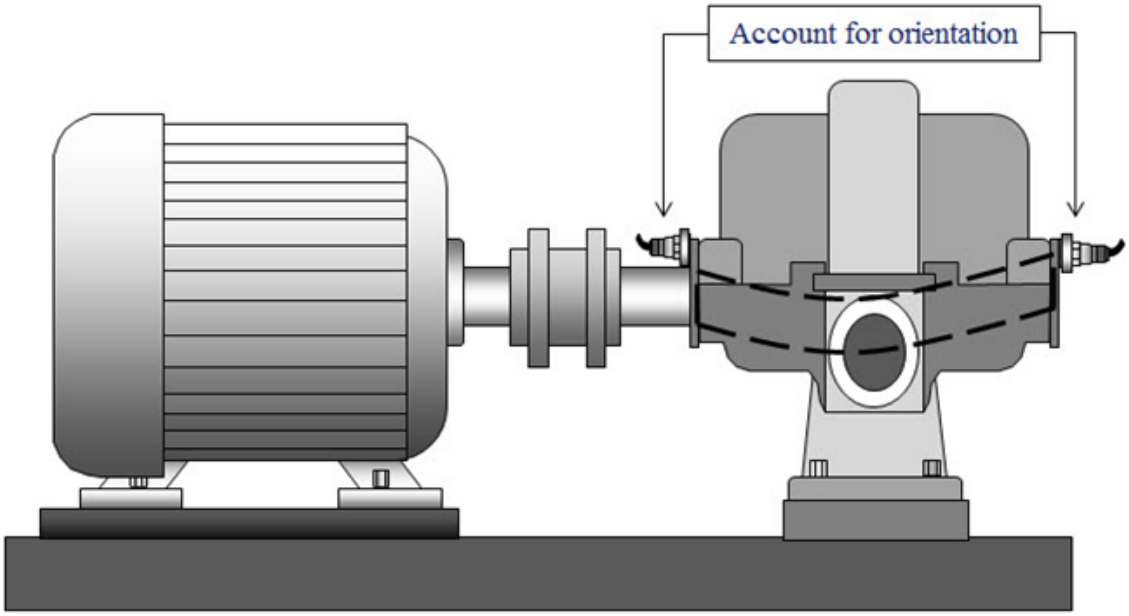
If the shaft is bent in the center, between the bearings, the phase reading will be 180 degrees across the bend. Absolute or relative phase reading may be close to zero degrees, due to sensor orientation. In the example given, the two sensors are already out of phase. Therefore the phase readings may be close to zero degrees.



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Soft Foot – Characteristics and Detection

Soft foot is a type of frame distortion usually caused by the mating of a flat machine foot to an uneven base or an uneven foot to a flat base. Tightening the fasteners between uneven surfaces can cause the frame to twist. In turn, the bearings and seals are no longer aligned.

Soft foot can cause excessive axial loading on the bearings and seals leading to accelerated failure rate. Monitor bearing temperature in addition to vibration. Bearing and seal temperatures tend to increase on a machine operating under a soft foot condition.

In some cases, stator related anomalies have been documented on electric motors.

Soft Foot Confirmation

Mount a dial indicator on the top of a foot. Depress it to force a slight deflection of the dial; .020 inches is usually enough. Loosen the fastener slightly, while watching the indicator. If the dial deflects in either direction, a soft foot is likely. However, this alone is not proof of soft foot.

Re-tighten the bolt until the condition can be resolved safely. Continue around the remainder of the feet to check for a worse condition elsewhere.

A running soft foot check can also be useful to verify soft foot. While monitoring the vibration in live mode, repeat the bolt loosening procedure as in the dial indicator method. If the amplitude decreases at any point, a soft foot is likely, however, it may not be soft foot.

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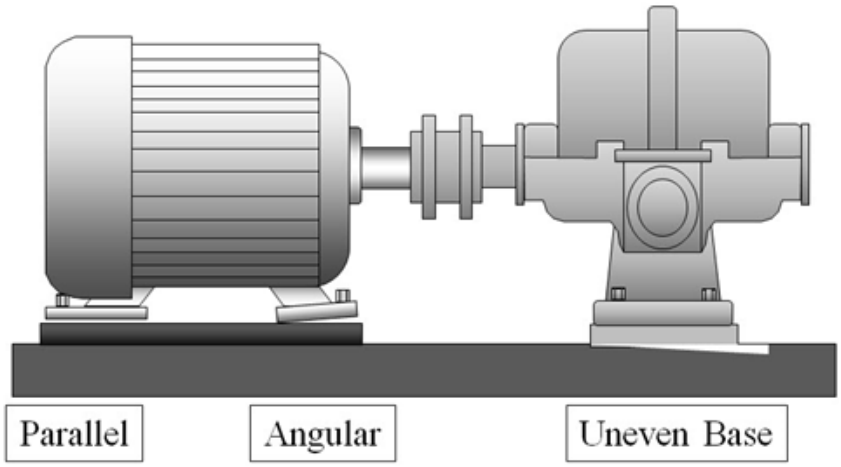
A machine can have as many soft feet as it has feet. That is, each foot may have some type of deformity or distortion that it constitutes a soft condition.

There are various types of soft foot. A foot may be bent up, down or to one side. One or more feet may be non-parallel with the rest of the feet.



Occasionally, the feet themselves may be in good condition, but the base the machine is mounted on is concaved, or convex.

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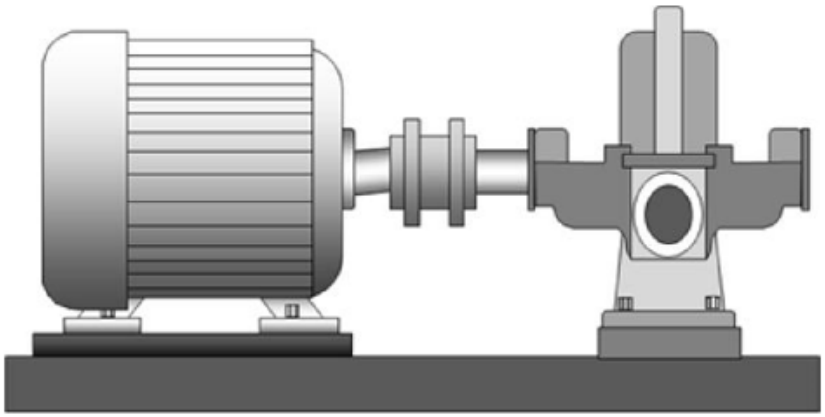
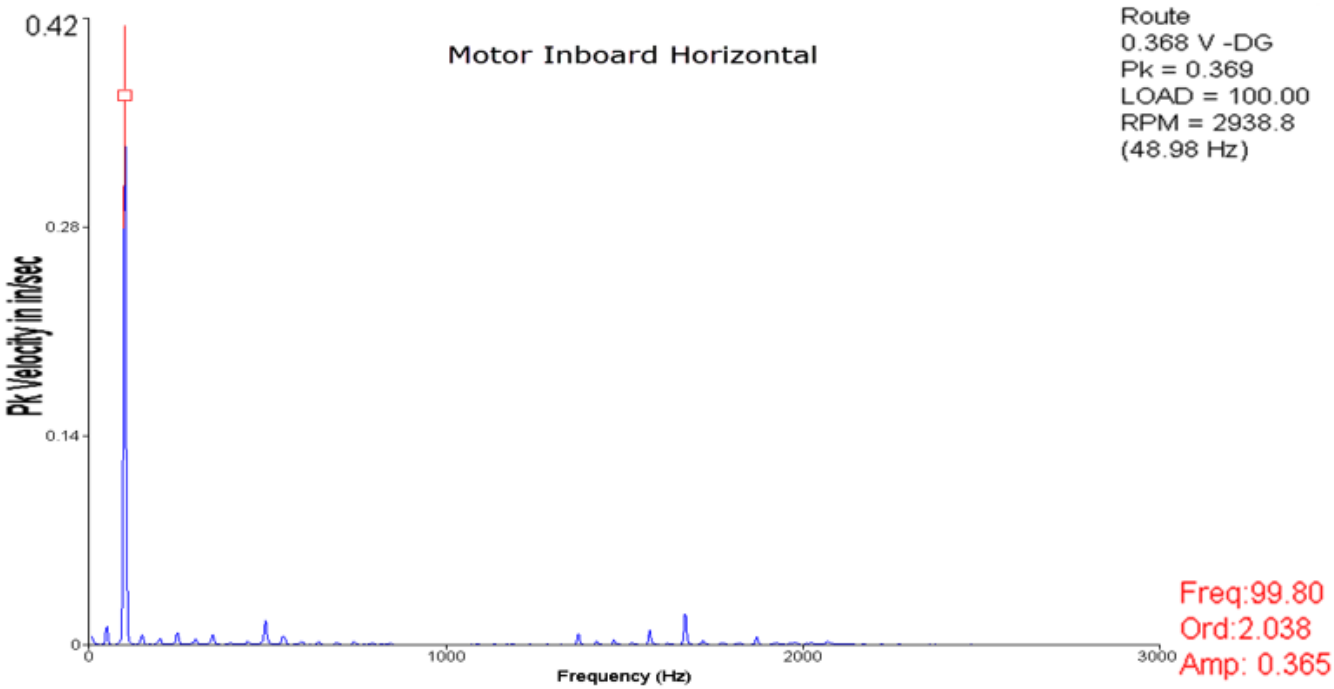
Soft Foot – Case History

- In-Line Coupled Center Hung Pump
- RPM = 2938
- Line frequency is 50 Hz



The spectrum has an overall amplitude of .36 in/sec measured from the motor inboard horizontal location. The dominant peak appears to be 2 x TS, which might lead the analyst to consider offset misalignment as a possible cause.



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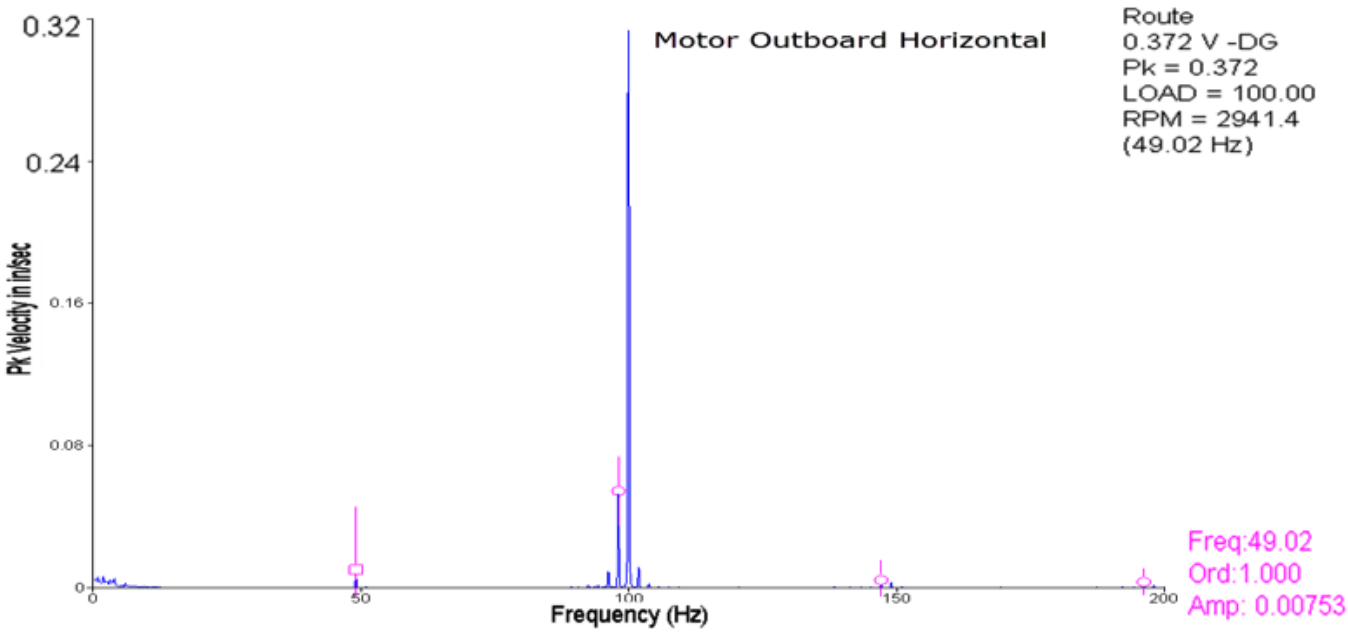
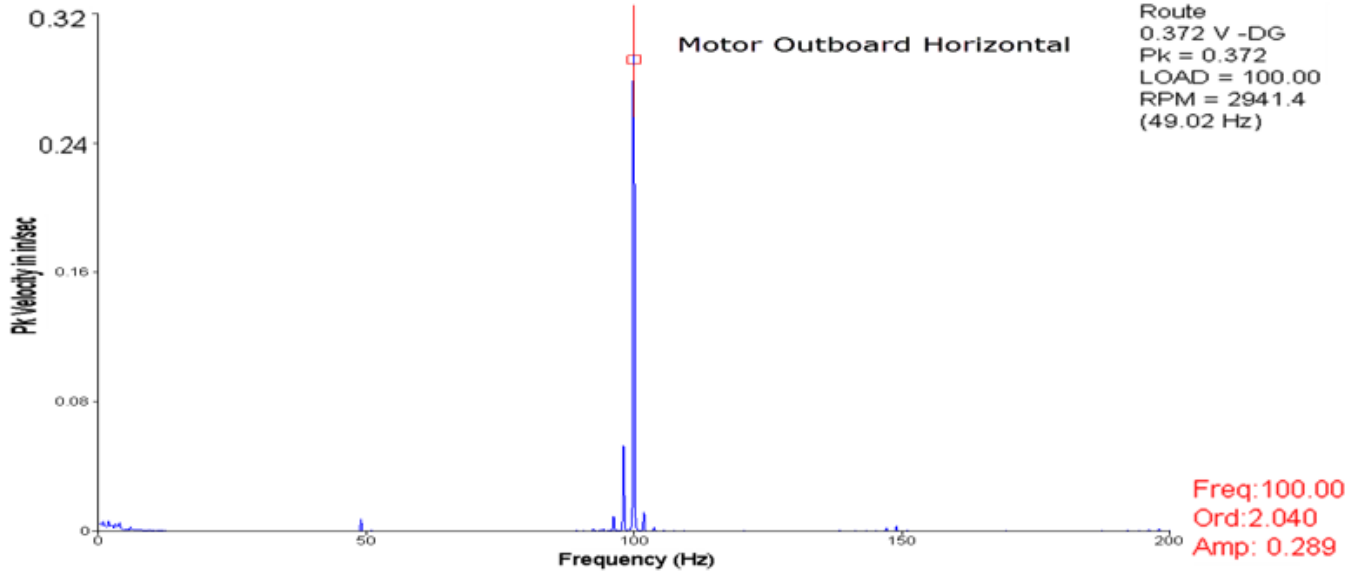
A higher resolution spectrum acquired at the motor outboard horizontal location shows that the dominant peak is not 2 x TS.



The dominant peak in the spectrum is actually at 100 Hz. This is 2 x Line Frequency (LF). The cause is very likely to be soft foot on the motor, which is causing distortion of the motor stator resulting in a high vibration amplitude at 2 x LF. Further testing should now be performed to verify a possible soft foot condition using the methods outlined previously in this section.



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Review Questions

1. Describe the phase relationship between a motor / pump combination with a flexible grid type coupling with offset misalignment.

2. Angular misalignment tends to appear in the spectrum at._____.

3. If a machine is operating under a combination of angular and parallel misalignment, what patterns would be expected in time and frequency data?

4. True or False: Rotating equipment having been aligned while static will still be aligned when it reaches normal operating parameters.

5. Bearing misalignment can be difficult to identify with only route vibration data. What else can be done to help identify a cocked bearing?

6. A general rule of thumb concerning misalignment states that if the amplitude of the 2 x TS peak is _____or more than the amplitude at 1 x TS, misalignment is present.

7. When the bend occurs closer to the end of the shaft, what will the spectral pattern be?

8. True or False: A precision balance job will cure a bent shaft condition.

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9. What are two tests that may be performed to verify a suspected soft foot condition?

10. A soft foot check should be performed prior to what corrective action?

11. True or False: Soft Foot is a type of looseness.

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PeakVue™

Section 7

Objectives

- Acquire a basic understanding of PeakVue Signal Processing.
- Learn to acquire and analyze PeakVue data.

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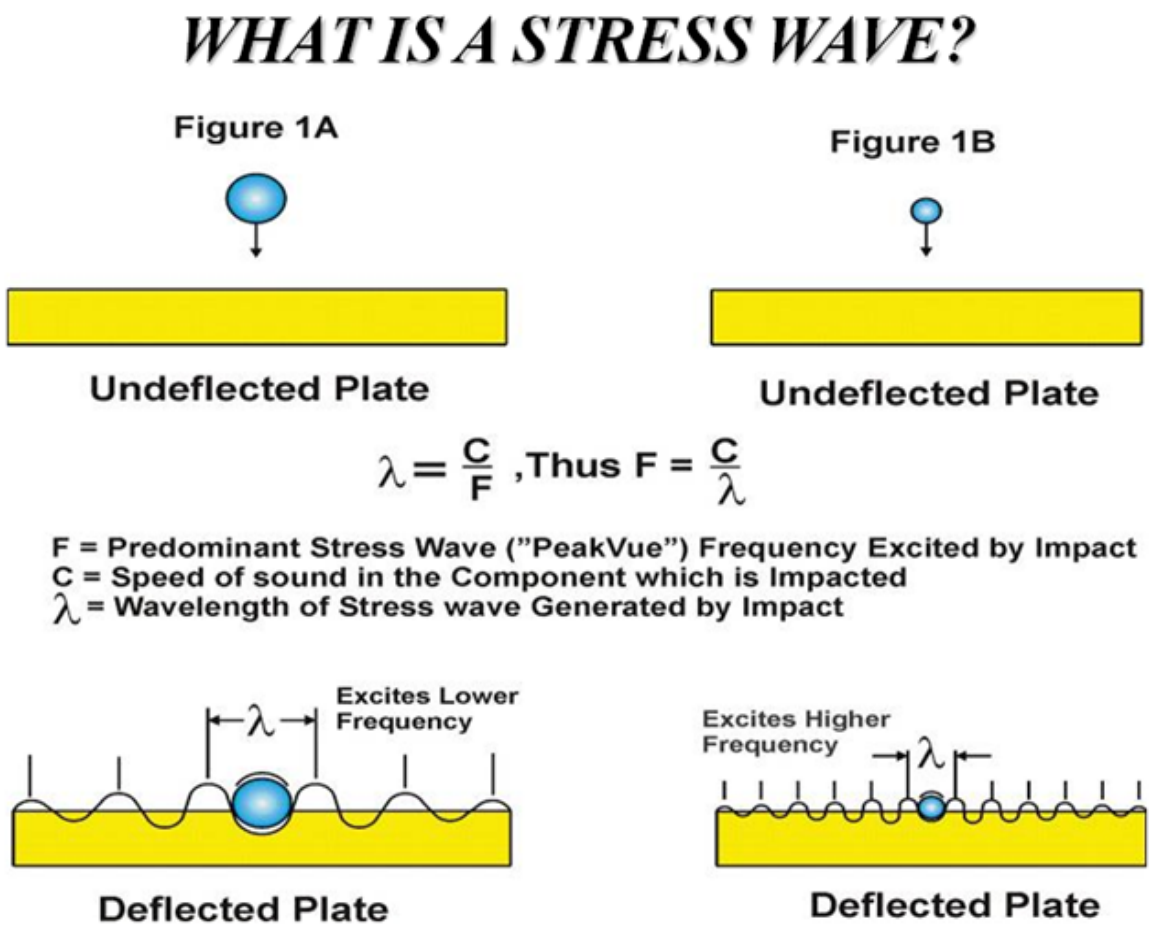
What is PeakVue?

PeakVue stands for Peak Value. PeakVue analysis is actually a measure of stress wave activity in a metallic component. Stress waves are associated with impact, friction, fatigue cracking, lubrication faults, etc., and are generated in various components, such as rolling element bearings and gears. For example, when a rolling element impacts a defect on a bearing raceway, it will generate a series of stress waves that propagate away from the location of the defect in numerous directions. The wave propagation introduces a ripple on the machine surface that causes a response input to a sensor that detects motion, such as an accelerometer.



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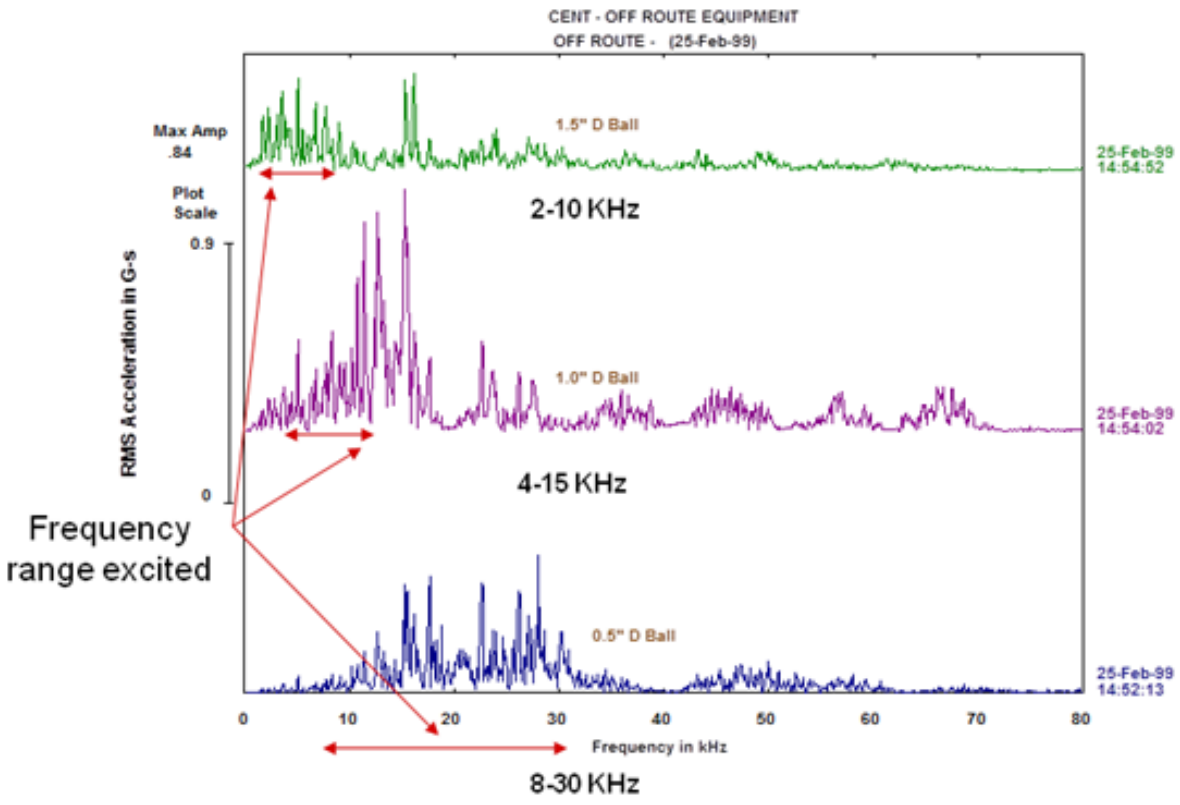
Stress Wave Characteristics

1. Short term transient events - microseconds to a few milliseconds in duration.

2. High frequency - generally concentrated from 1kHz to 15 kHz.

The frequencies generated by the stress waves are predominantly controlled by the ratio of the speed of sound within the media over the wavelength. Frequencies are largely concentrated in the 1 to 15 kHz range (largely dependent on the mass and geometry of the impacting object, the type of surface it impacts, etc.) Stress wave frequencies can extend up to 50 KHz. Stress waves also excite and include frequencies excited by system resonances. However, it is surprising that the contribution of such resonant responses is typically only five to 10 percent of total stress wave content.



Frequencies excited by impacting of 0.5”, 1.0” and 1.5” diameter balls.

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The time duration of the stress wave event will be dependent upon:

1. Type of event. (e.g., stress waves from impacting will last longer than stress waves accompanying the release of residual stress buildup through fatigue cracking)

2. Relative location of the sensor (accelerometer) to the initiation site.

3. Severity of the fault responsible for the stress wave emission.

4. Material the stress waves are traveling through.

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Sensor Mounting

PeakVue's ability to detect various fault types is determined by both the transducer mounting surface and the mounting method. The variations that exist in each application can limit the Fmax and high-pass filter that can be used in a PeakVue measurement (painted versus unpainted surface, flat versus curved surface, smooth versus rough surfaces, etc.).

A general purpose 100 mV accelerometer, mounted with a 2-pole magnet has been found to be useful for stress wave detection in most applications. Painted surfaces should be avoided. Thick paint attenuates stress waves. There should be a minimum of dual line contact made between the magnet rails and the curved surface of the machine. Limitations on using this mounting scheme will be addressed later in this chapter.

A flat, rare earth magnet will capture significantly more stress wave amplitude than will a 2-pole magnet, if mounted on a flat, reasonably clean surface. This is particularly true when a frequency bandwidth (Fmax) greater than approximately 2000 Hz is needed or when a high-pass filter greater than 2000 Hz is used. Tests have shown that if either of these two conditions exists, stress waves above approximately 3000 Hz, which are detected by a flat rare earth magnet, can be missed altogether by a 2-pole magnet when making PeakVue measurements. Use of the hand-held stinger type probe is not recommended.

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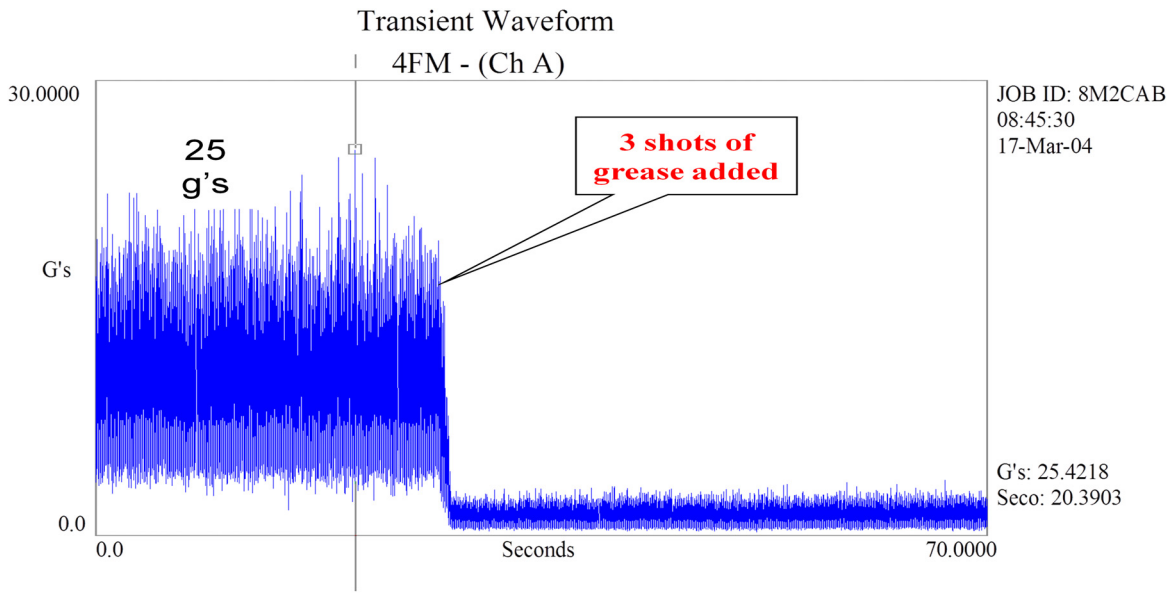
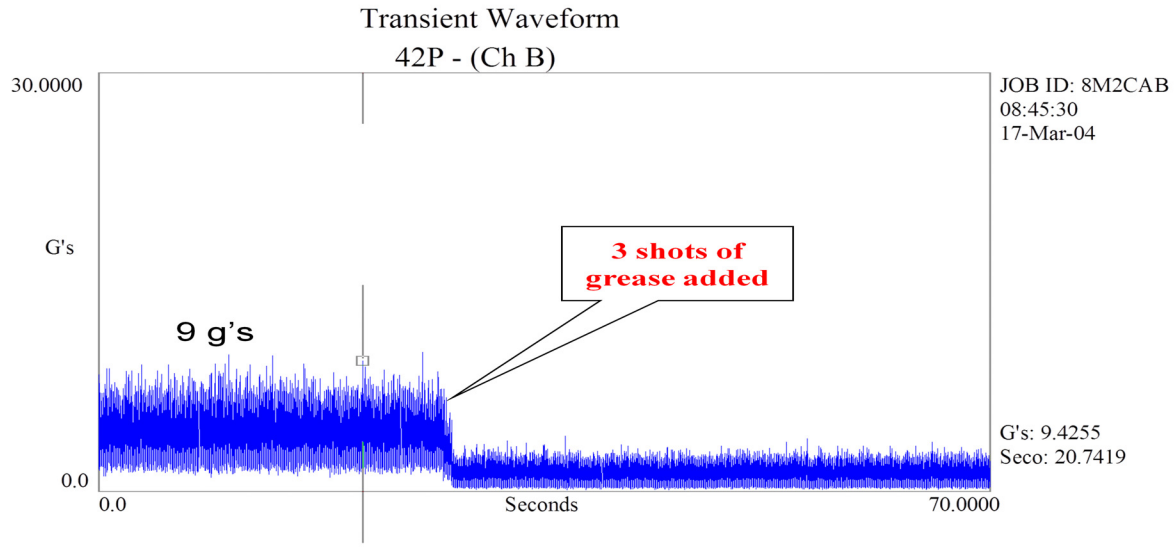


Channel A of 2 channel transient waveform on a bearing with a lube fault with flat magnet on sensor pad.

Stress waves attributed specifically to fatigue and friction typically excite frequencies between 10 and 50 kHz. For this reason a flat, rare earth magnet is the preferred method of mounting when dealing with lubrication issues. While both conditions were identifiable in this comparison data, the data taken with the flat rare earth magnet is obviously more conclusive.



Channel B taken with two-pole magnet.

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PeakVue Signal Processing

PeakVue signal processing differs from normal signal processing in many ways.

1. **Filtering** - PeakVue uses a high pass filter instead of a low pass filter. The high pass filter is used to separate the macro vibrations (or normal vibrations) from the micro vibrations (Stress waves).
2. **Sample Rate** - PeakVue always uses a fixed sample rate (102,400 Hz). In normal data collection, the sample rate is dependent upon the spectral Fmax (2.56 x Fmax).
3. **Peak Value** - The PeakVue waveform stores peak amplitude values, making trending and severity determination possible.
4. **Full Wave Rectification** - The PeakVue waveform is a display of zero to peak values over time.
5. **Averaging** - Only one average of the waveform is used to compute the PeakVue spectrum.

Filtering

Although band pass and notch filters are available for use under special circumstances, PeakVue normally uses a high pass filter, set either at or above the Fmax, to filter out macro vibrations. This is different from conventional data collection, where a low pass filter is set at the Fmax to prevent aliasing. The use of the high pass filter allows the user to filter out all of the normal (or macro) vibration, allowing for the separation and analysis of only the stress wave (micro) vibration.

Sample Rate

The digital sample rate utilized for PeakVue is always 102,400 Hz (samples per second). This sample rate is fixed no matter the Fmax setting. It allows for the capture and analysis of very high frequency stress waves, even when the Fmax is set very low. This is especially useful in the analysis of slow speed equipment. When conventional data collection techniques are utilized, the digital sample rate decreases when the Fmax is lowered.

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Peak Value

For PeakVue, the waveform holds the highest amplitude sampled in a period that is determined by calculating the inverse of the Fmax multiplied by 2.56. This is accomplished by absolute peak decimation. This is what makes PeakVue a much better tool than simple demodulation for trending and evaluating the true severity of the problem. To get a better understanding of how this works, compare a normal waveform to a PeakVue waveform.

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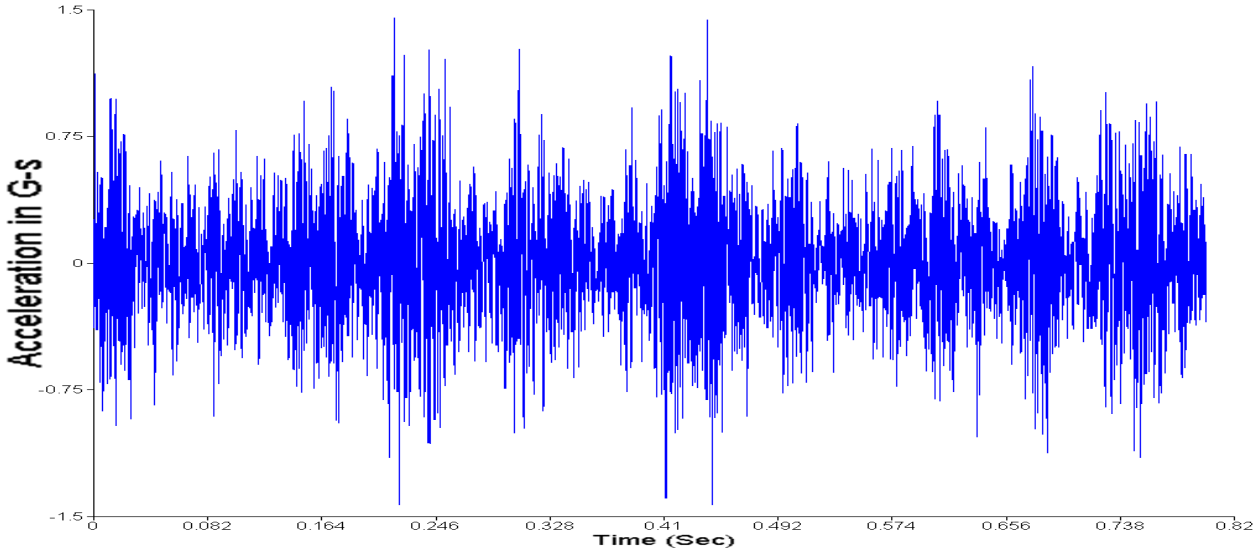
Normal (Conventional) Waveform

The normal waveform is sampled at a rate that is always 2.56 times the Fmax. For example, using a 1000 Hz Fmax along with 800 lines of resolution, the resultant waveform would be .8 seconds long and contain exactly 2048 data points.



Sample Rate = 2560 Hz
Waveform time length = .8 sec
Total data points = 2048

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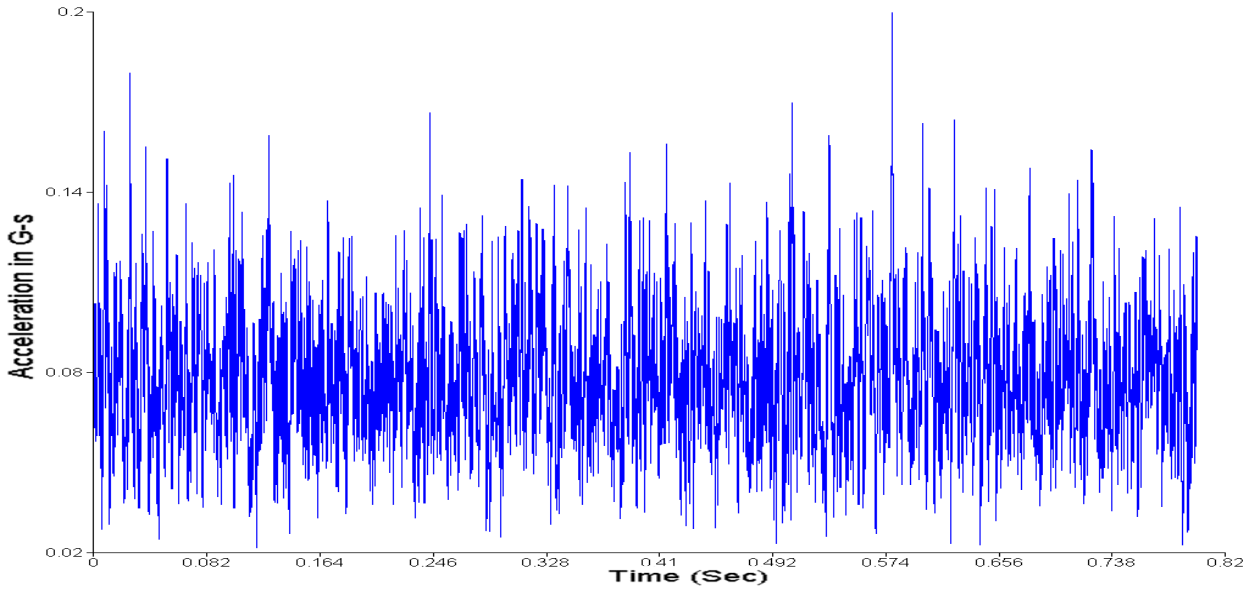
PeakVue Waveform

The PeakVue waveform is sampled at a rate of 102,400 Hz. However, peak decimation occurs over a time period that is equal to the normal sample period. So using a 1000 Hz Fmax along with 800 lines of resolution, the resultant waveform would still be .8 seconds long and contain exactly 2048 data points.



Sample Rate = 102,400 Hz
Waveform time length = .8 sec
Total data points = 2048

Notice that in this example, the PeakVue sample rate is exactly 40 times faster than what the normal sample rate would be for a 1,000 Hz Fmax. In this example, only one value out of 40 is actually being stored to the waveform. This stored value is the **peak value** that occurs during the period of 40 samples. The fixed sample rate used by PeakVue is comparable to what the normal sample rate would be if the Fmax was set to 40 kHz.

(40,000 x 2.56 = 102,400)

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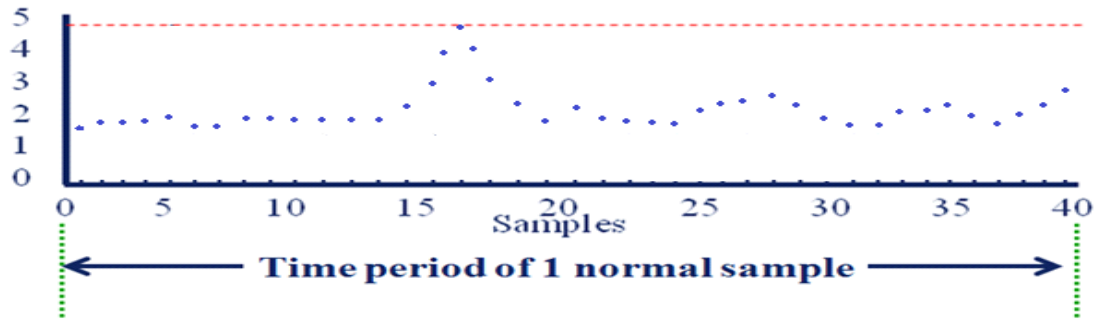
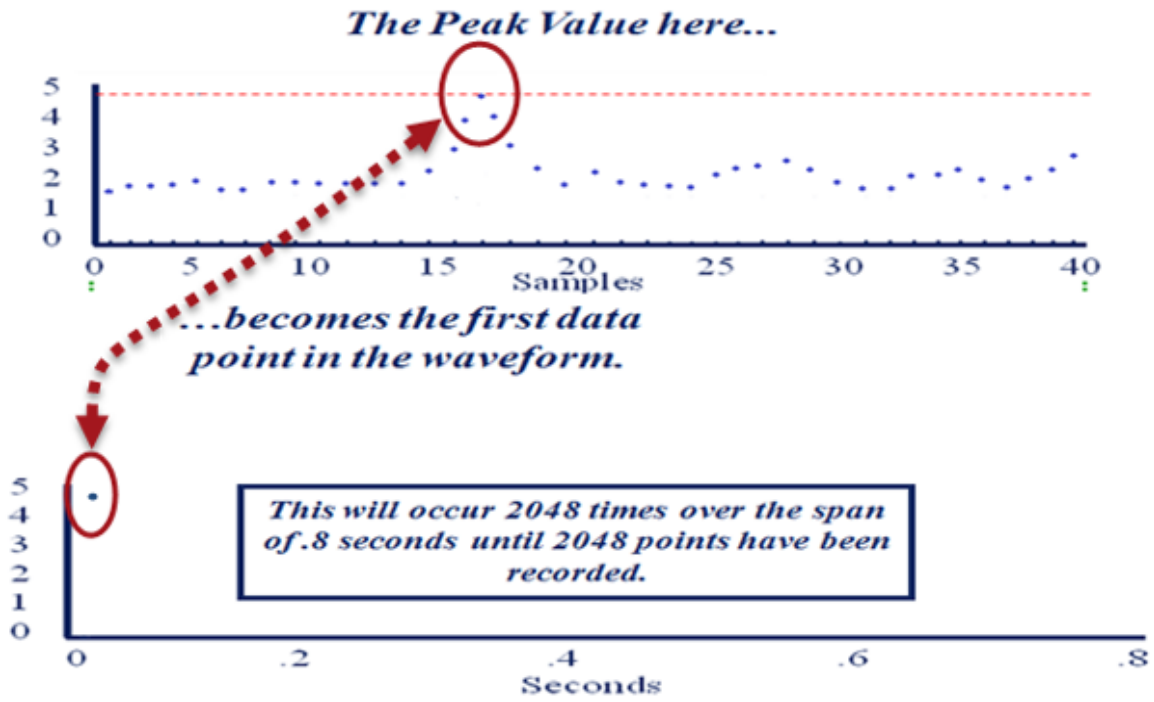


The peak value that is measured over the span of these 40 samples, then becomes the first data point in the PeakVue waveform.



If the PeakVue Fmax is raised to 2000 Hz, the sample rate remains the same. However, the rate at which peak values are stored to the waveform will now double to 5,120 Hz. Since the PeakVue sample rate is now only 20 times faster than the normal sample rate, Peak Decimation will now occur over the span of 20 samples instead of 40. For PeakVue, raising the Fmax increases the rate at which peak values are stored to the waveform, but not the rate at which the actual sampling occurs. Once the waveform is completed, the analyzer then performs FFT on the PeakVue waveform to generate the PeakVue spectrum.

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Rules for PeakVue Data Acquisition

PeakVue is a very powerful data capture and analysis technique. However, certain rules and recommendations should be followed to make sure that the acquired data is both valid and meaningful. These rules may change according to the speed and the type of component being measured.

- For an antifriction bearing application, the Fmax should be four times the highest calculated defect frequency. If the bearing fault frequencies are not known, then it is common to use an Fmax of approximately 40 x turning speed.
- For a gearbox, the Fmax should be at least higher than 2 x the gear mesh frequency. (usually around 2.25 x GMF)
- Use enough lines of resolution to ensure that the waveform is long enough to contain at least 15 revolutions of the shaft.
- Only one waveform average is used for the PeakVue spectrum. Make sure to set the number of waveform averages to one.
- The high pass filter is set at or above the Fmax. If there is no available high pass filter at the specified Fmax, then use the next highest available high pass filter past the Fmax.
- PeakVue is always measured with an accelerometer and the data is always stored in acceleration units and analyzed in acceleration units. Since there is no integration being performed, the **Obtain Special Time Waveform option in the measurement point setup should be disabled**.

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Available PeakVue Fmax Values (2130)

1Hz	**20Hz**	**100Hz**	**500Hz**	**2500Hz**
2Hz	**25Hz**	**125Hz**	**625Hz**	**4000Hz**
4Hz	**32hz**	**160Hz**	**800Hz**	**5000Hz**
5Hz	**40Hz**	**200Hz**	**1000Hz**	**8000Hz**
8Hz	**50Hz**	**250Hz**	**1250Hz**	**10000Hz**
10Hz	**64Hz**	**320Hz**	**1600Hz**	
16Hz	**80Hz**	**400Hz**	**2000Hz**	

Available PeakVue Filters

500Hz HP	**20-150Hz BP**	**50Hz Notch**
1000Hz HP	**50-300Hz BP**	**60Hz Notch**
2000Hz HP	**100-600Hz BP**	**100Hz LF Rej**
5000Hz HP	**500-1000Hz BP**	
10000Hz HP	**5-6.5kHz BP**	
20000Hz HP		

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Example PeakVue Setup #1

Motor driven pump with antifriction bearings
Bearing Number - Unknown
1780 RPM

The first consideration is usually the Fmax. According to the previous rule, the Fmax needs to be 4x the highest calculated defect frequency, which would be the BPFI. Since the BPFI is unknown in this case, we will use an Fmax of 40 x RPM. This is about 1186 Hz (40 x 1780/ 60). By looking at the previous page, 1186 Hz is not an available PeakVue Fmax. So, the Fmax would actually be 1250 Hz. Use of a 1250 Hz Fmax would then force the use of a 2000 Hz HP filter. In this case, it may be better to round down a little and use a 1000 Hz Fmax so that the 1000 Hz HP filter could then be used. **The result would be a 1000 Hz Fmax with a 1000 Hz HP filter.**

The next consideration for the data acquisition setup is for the lines of resolution. As was mentioned earlier, the determining factor for the number of lines is the waveform length. The time in the waveform needs to be long enough to contain at least 15 shaft revolutions. An easy way to determine whether or not the waveform is long enough, is to calculate the waveform time length using the **# lines ÷ Fmax** formula. Only in this case, start with the Fmax in orders, so that the waveform time length will then be calculated in orders.

To convert the Fmax into orders, simply divide Fmax in Hz by the turning speed in Hz. In this case:

1780 RPM ÷ 60 = 29.7 Hz. Then **1000 Hz ÷ 29.7 = 33.7**

The Fmax in orders is **33.7**.

Suppose the number of lines of resolution is set to 800. Use 800 and divide by 33.7 to get 23.7. The waveform will have 23.7 shaft revolutions, which satisfies the 15 revolution requirement.

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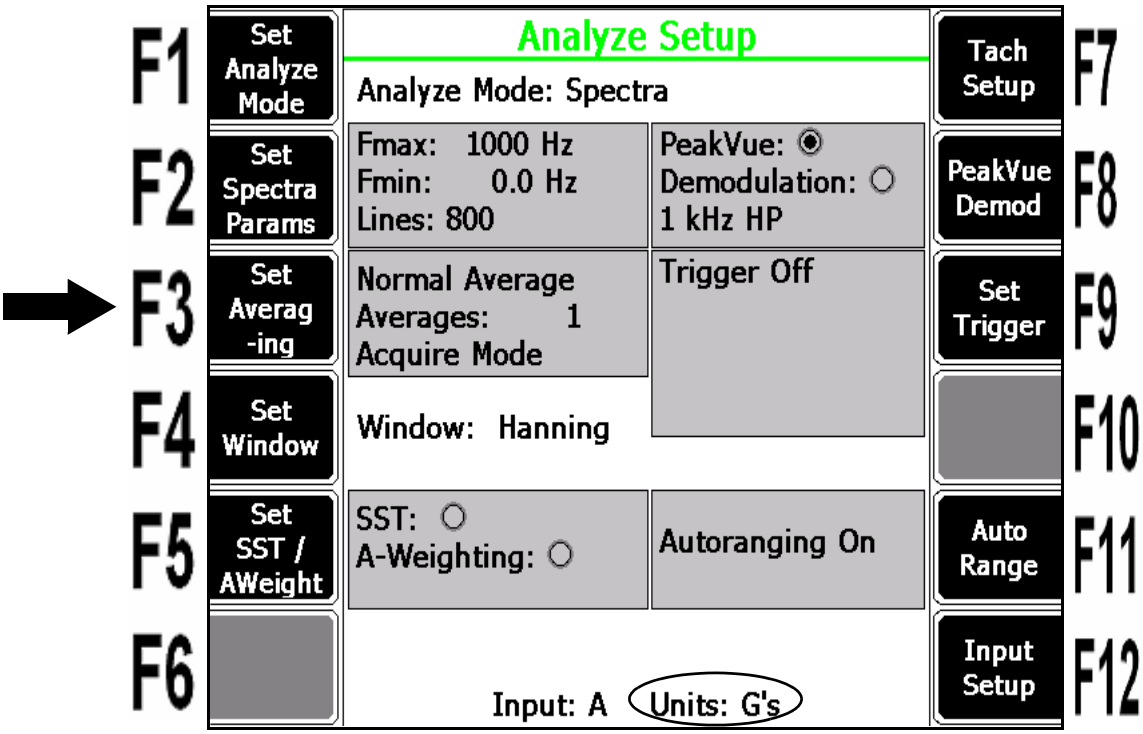

The resulting set up would look like this on the CSI 2130. Remember to take only one average, and use G's as the data units. Since there is no integration being performed, the integration setting will have no effect on the data.

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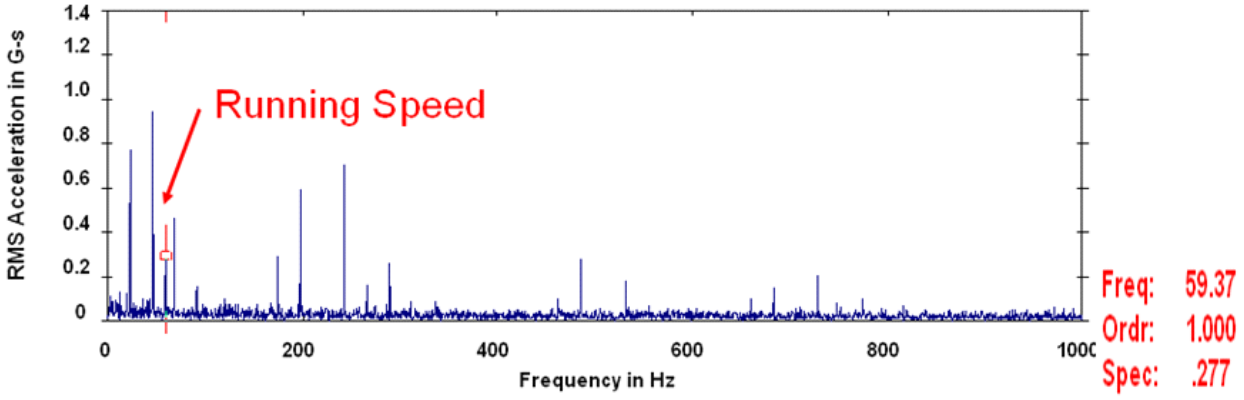
PeakVue Analysis

In addition to understanding how PeakVue works and how to acquire PeakVue data, an analyst needs to know how to analyze the PeakVue data. The analysis can be relatively simple provided the analyst follows a few basic steps:

First: Use the spectrum to identify the cause of the problem. The peaks in the spectrum indicate periodic modulations of stress waves and not frequencies of the waves themselves. Remember, either a high pass or band pass filter has removed any vibration frequencies below the Fmax.

Second: Always analyze the waveform for the determination of severity. The thing that sets PeakVue apart from simple demodulation is its use of peak values. This makes the waveform amplitudes trendable and valid for severity determination.

Step 1: Analyze the spectrum for any obvious frequency content.



Note

The peaks in the PeakVue spectrum represent **modulation frequencies** of high frequency stress waves and not actual vibrations at those indicated frequency.

Consider the peak at 1 x turning speed in this spectrum. It does not represent a vibration with a period that corresponds to 1 x turning speed. It represents is a recurrence of high frequency stress waves once per revolution of the shaft.

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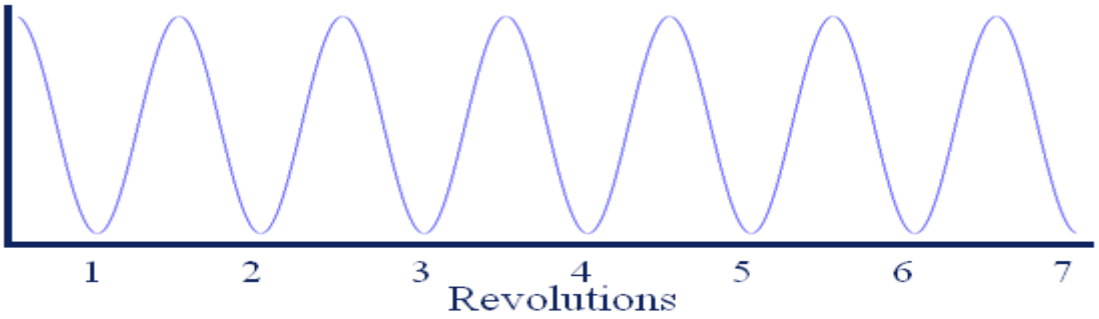
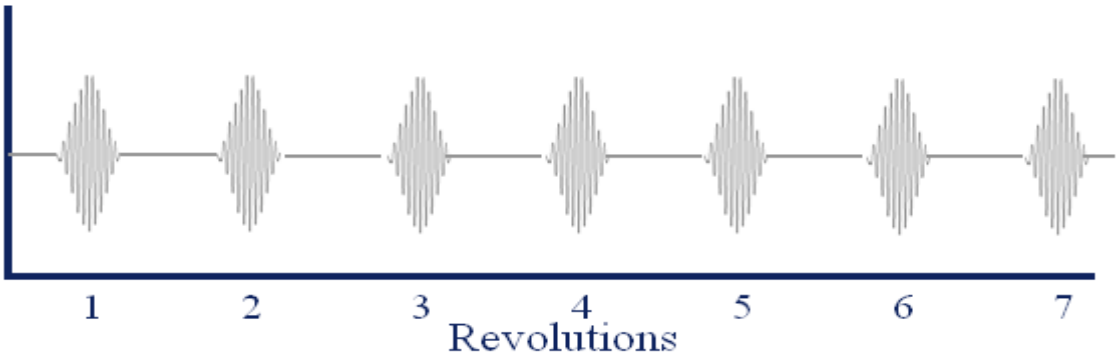


Vibration at 1 x turning speed.



Stress waves repeating at 1 x turning speed.

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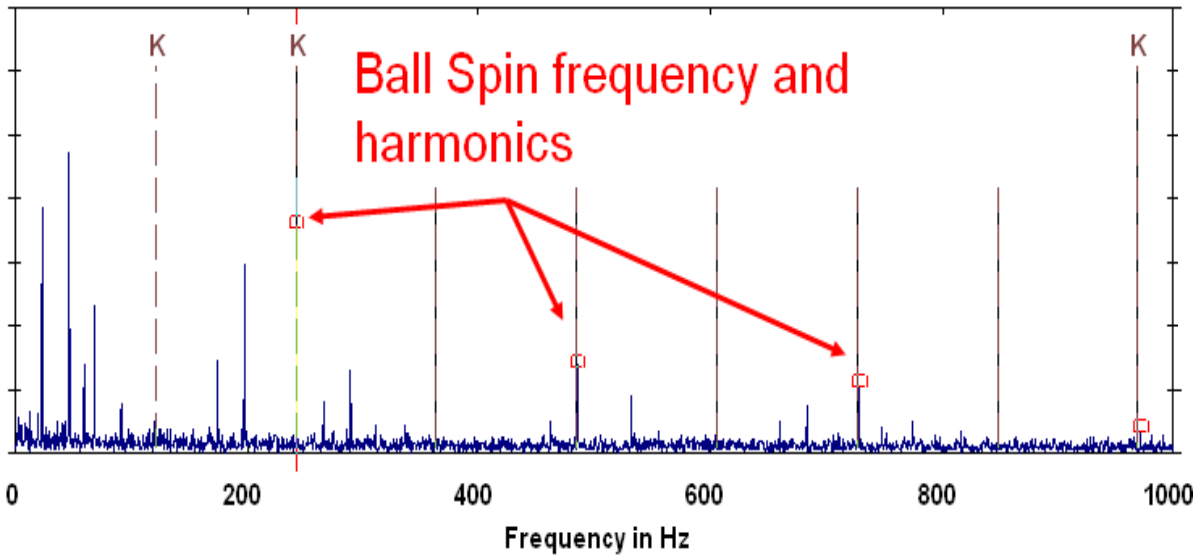
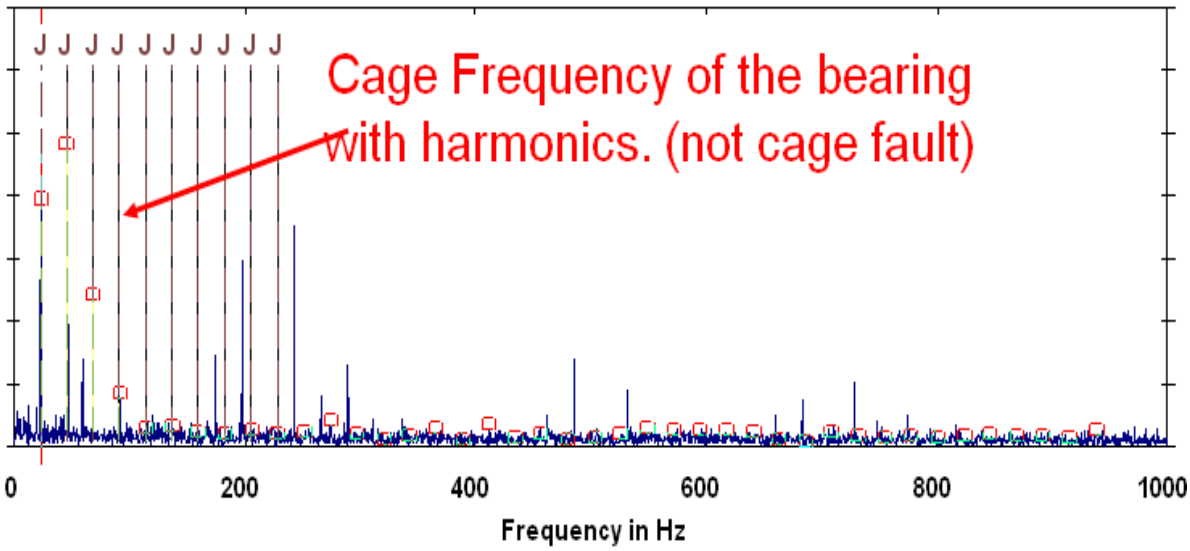
The additional peaks in the spectrum are all indications of a roller defect in the bearing.





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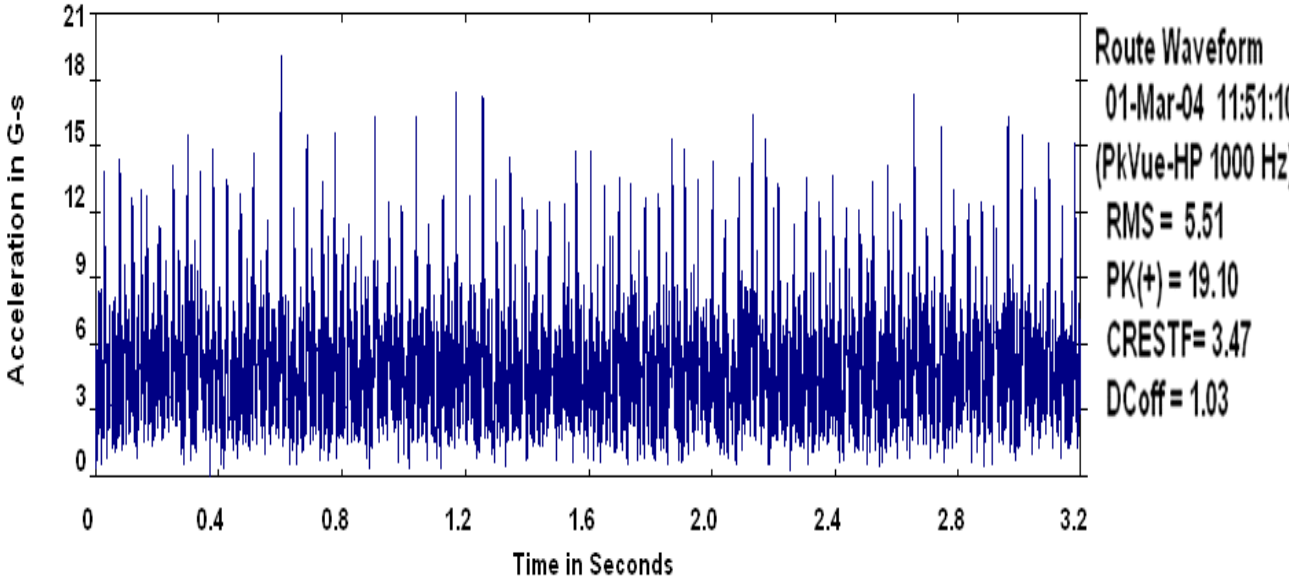
Step 2: Analyze the waveform to determine the severity.



The waveform is really the only source of amplitude and severity information. Therefore, any analysis of PeakVue data must include waveform analysis. The waveform amplitude value that is of most value is the **Max Peak waveform** value. The Max Peak value is simply the amplitude of the largest peak in the waveform. In this example, the Max Peak amplitude is 19.1 G's.

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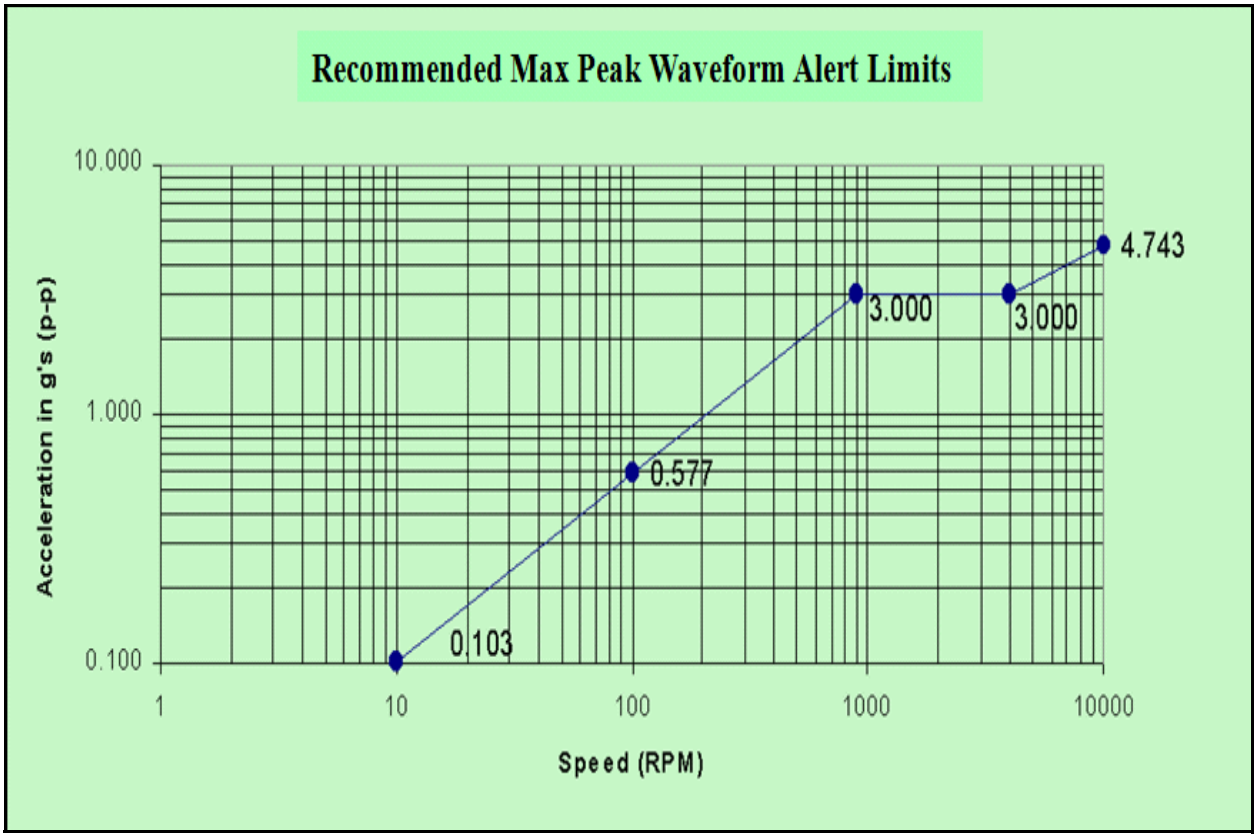
Recommended PeakVue Alert Levels

Recommended alert and fault levels for the Max Peak value in the waveform may vary significantly, depending on the type of defect, and on the RPM of the equipment. Considerations should also be made for the type of magnet used, the location of the data collection point in reference to the fault location, the criticality of the machine and the size of the machine. It is the trend of the max Peak value that is more useful in the severity assessment than the actual amplitude on any one day. The following charts are intended to give the analyst a rough idea of which amplitudes are acceptable in different situations, and which ones are not. The values listed are for recommended alert levels. To calculate the fault levels, multiply the alert level by two.



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COMPONENT RPM	R.E. BEARING FAULTS		GEAR FAULTS	
	Inner Race, Cage or Rolling Element Fault	Outer Race Fault	Worn or Scored Teeth[3]	Cracked Teeth[4] (Fully Loaded)
0-900	$\text{Nominal Speed Alarm } X \left(\frac{\text{Actual RPM}}{900} \right)^{0.75}$			
901-4000 (Nominal Speed)	3g	6g	3g	6g
4001-10,000	$\text{Nominal Speed Alarm } X \left(\frac{\text{Actual RPM}}{4000} \right)^{0.5}$			
10,001-UP	5g	10g	5g	10g

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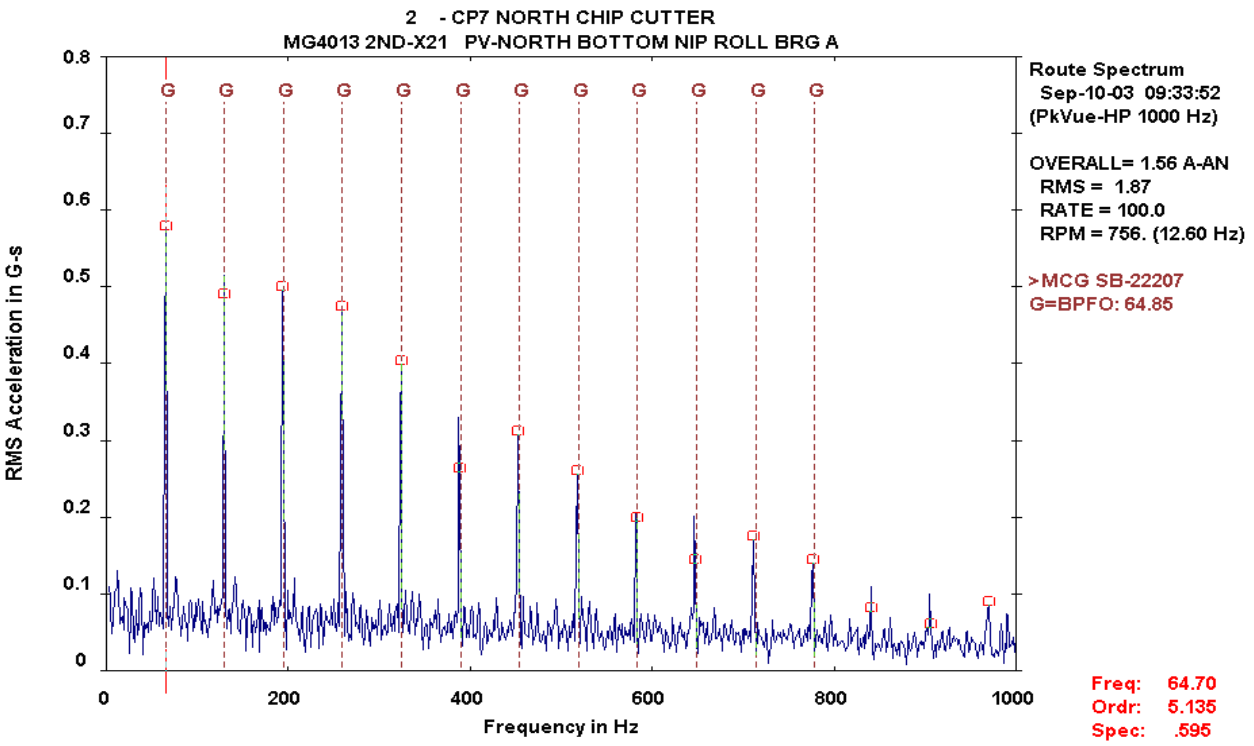
PeakVue Fault Identification

The following PeakVue case histories offer good examples of what to look for when analyzing PeakVue data. The most common types of faults are antifriction bearing defects. PeakVue is valuable for identifying any type of problem where metal to metal contact is occurring such as fatigue, looseness, gear defects and lubrication defects.

BPFO



The typical outer race defect will show up as a peak (often along with harmonics) at the bearing's calculated BPFO.

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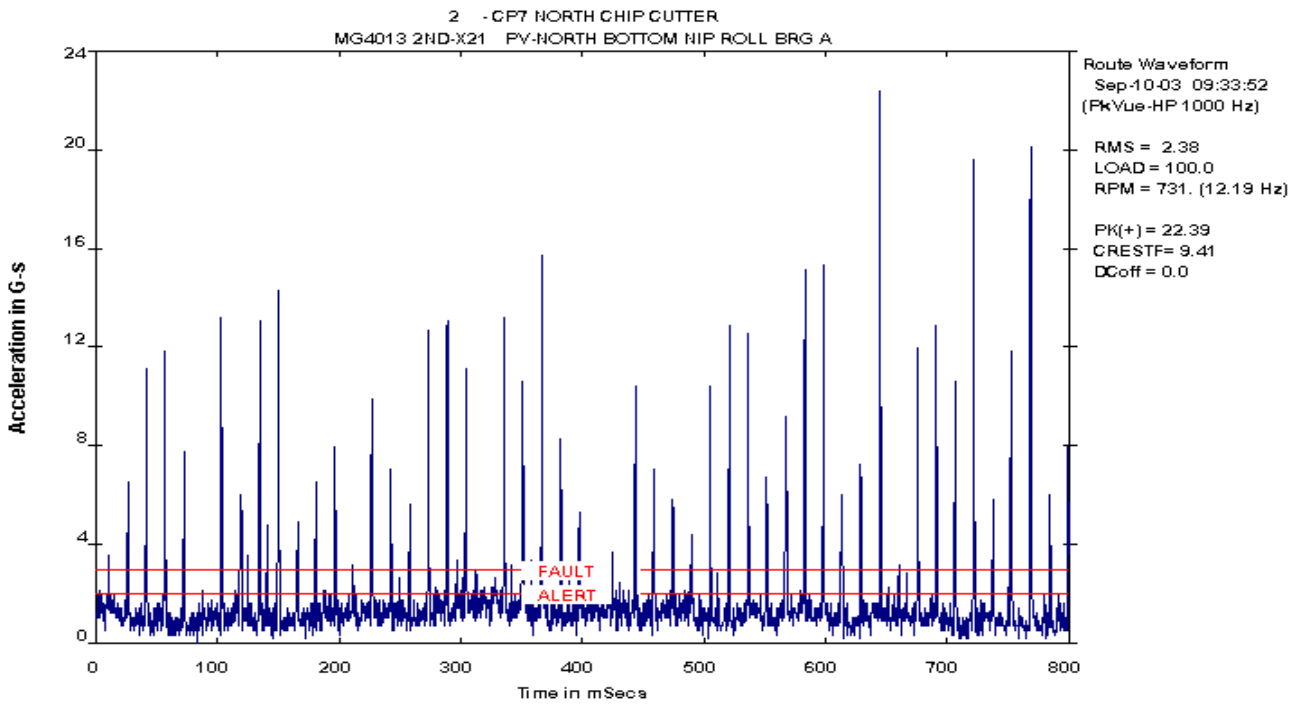

The waveform amplitude is 22 G's, indicating that it's time to replace this bearing. A picture of the defective bearing reveals that the damage to the outer race was very severe.

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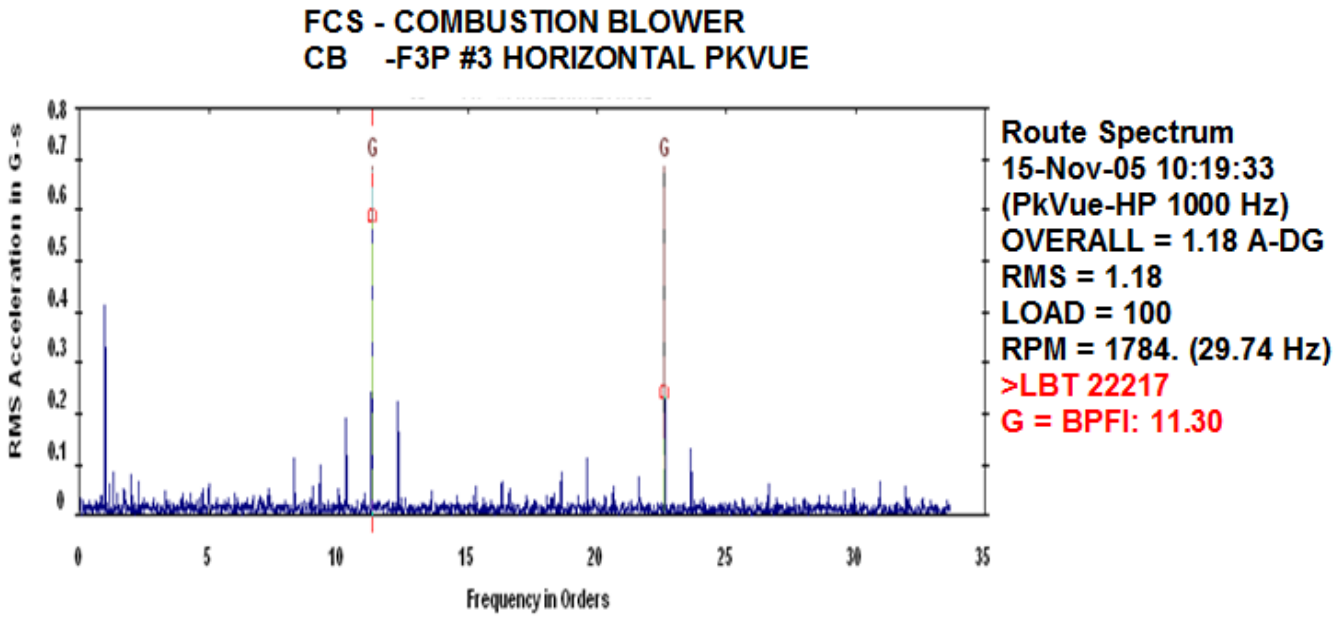
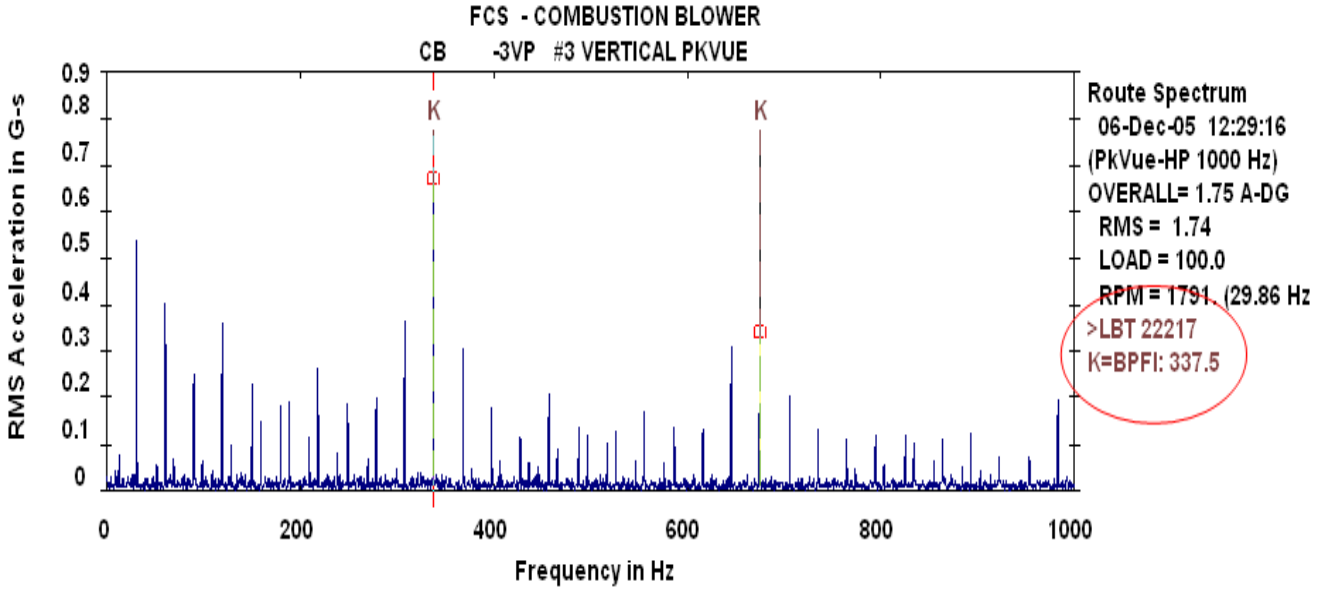
BPFI



Typical inner race defect shows peaks at calculated BPFI. Harmonics of BPFI are also common.



This is data taken the following month. Notice the additional peaks now present in the spectrum.

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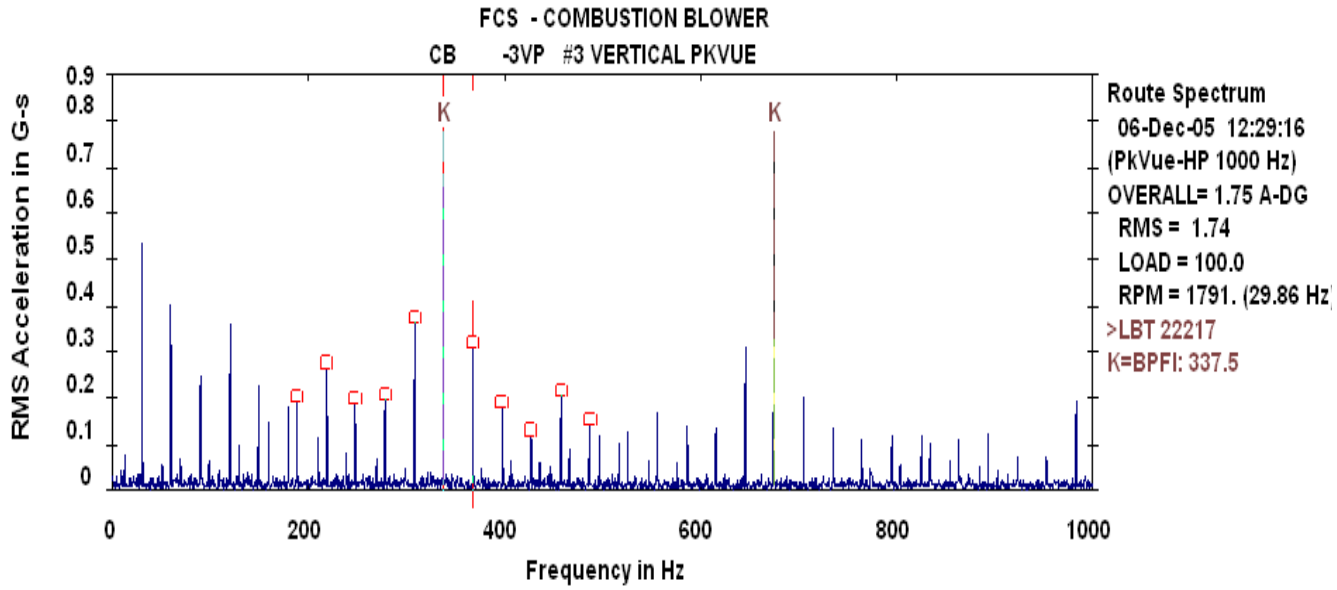
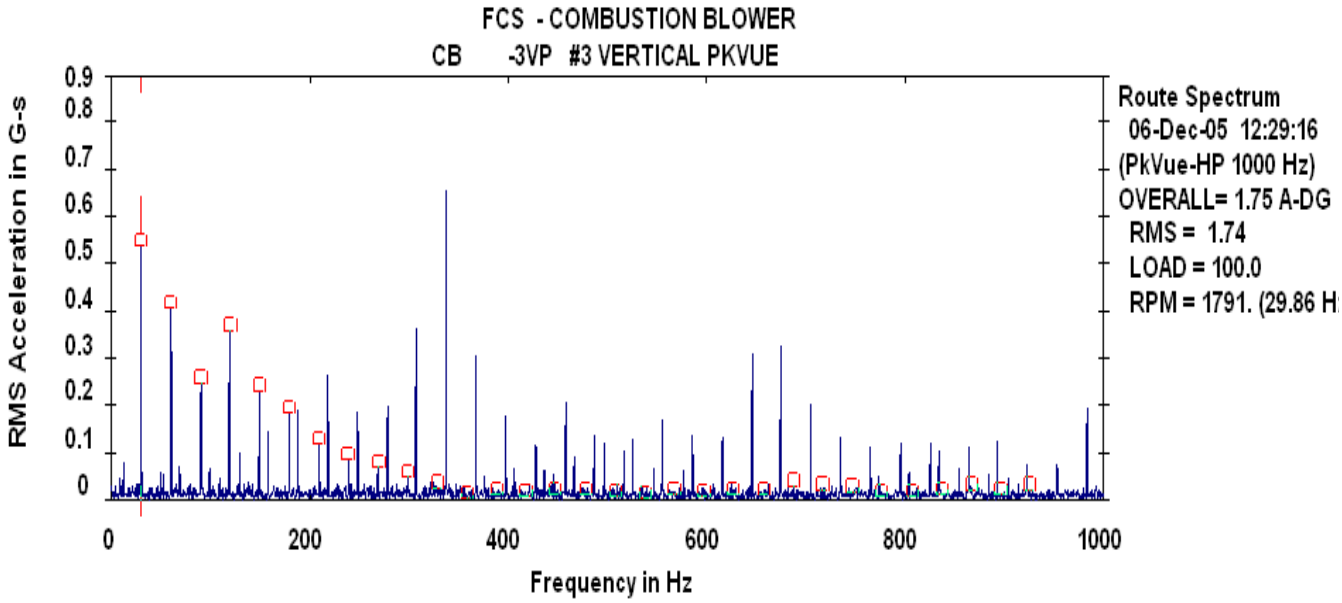


Turning speed sidebands are present around both harmonics of BPFI.



Turning speed and harmonics of turning speed are also present. It is most likely that the turning speed peaks and sidebands are not present as a result of looseness. They are modulation frequencies caused by the inner race defect going in and out of the load zone of the bearing, once per revolution of the shaft.

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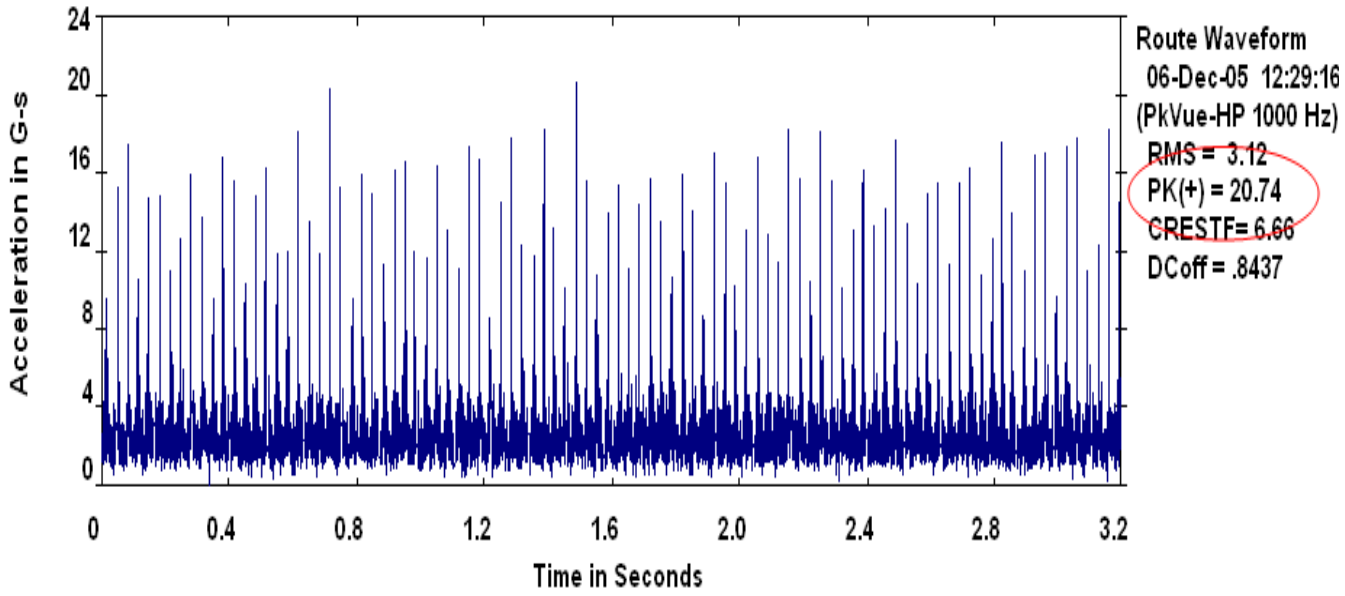

The waveform amplitude is 20.74 G's. According to the previous reference charts, this bearing needs to be scheduled for replacement.

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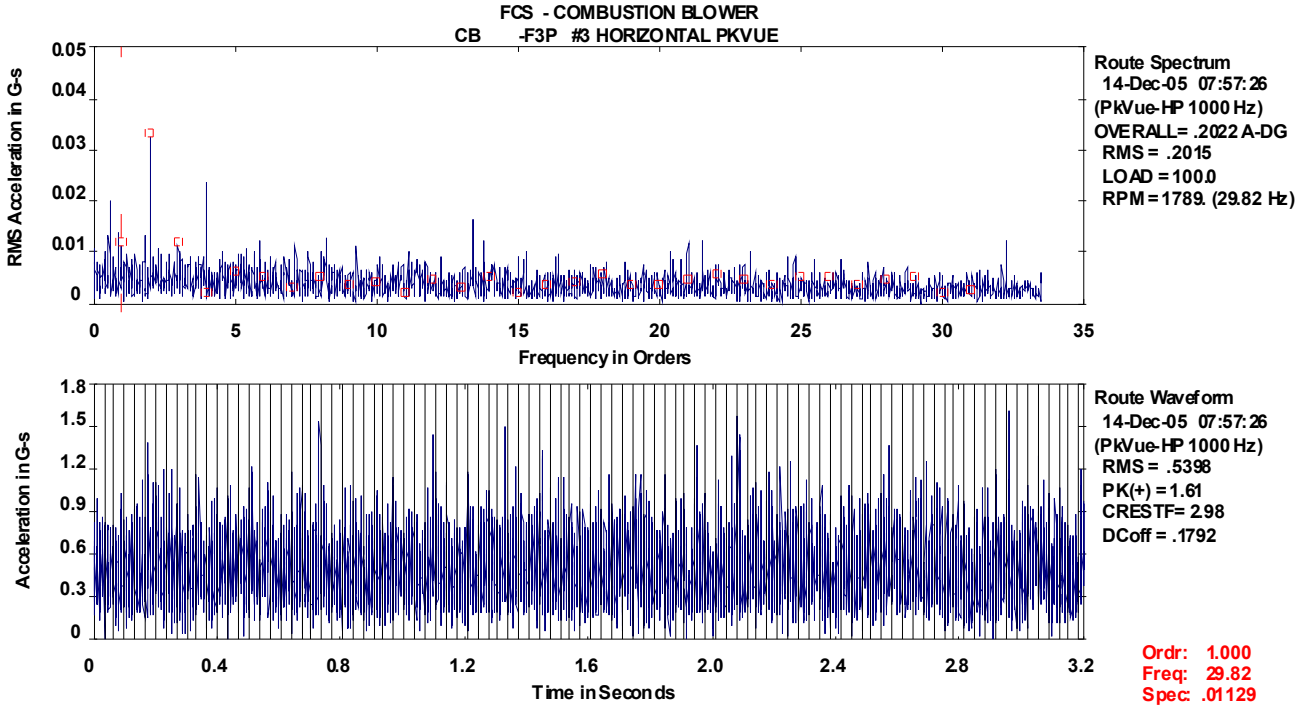
Data acquired after the bearing replacement.



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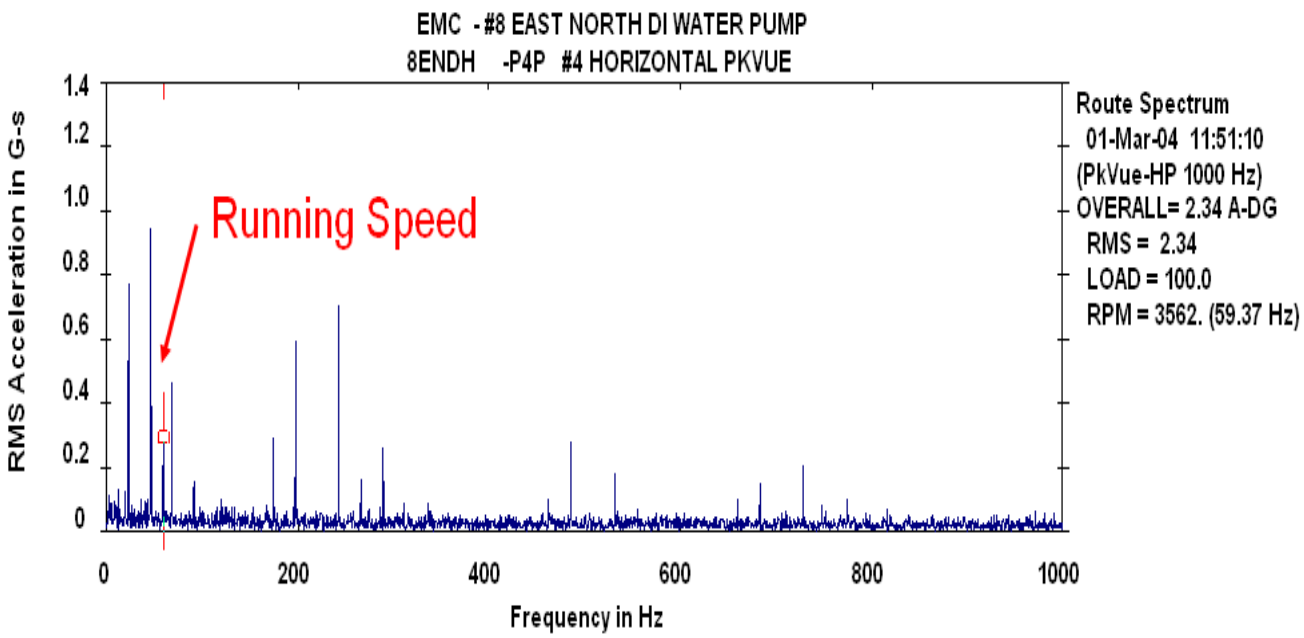
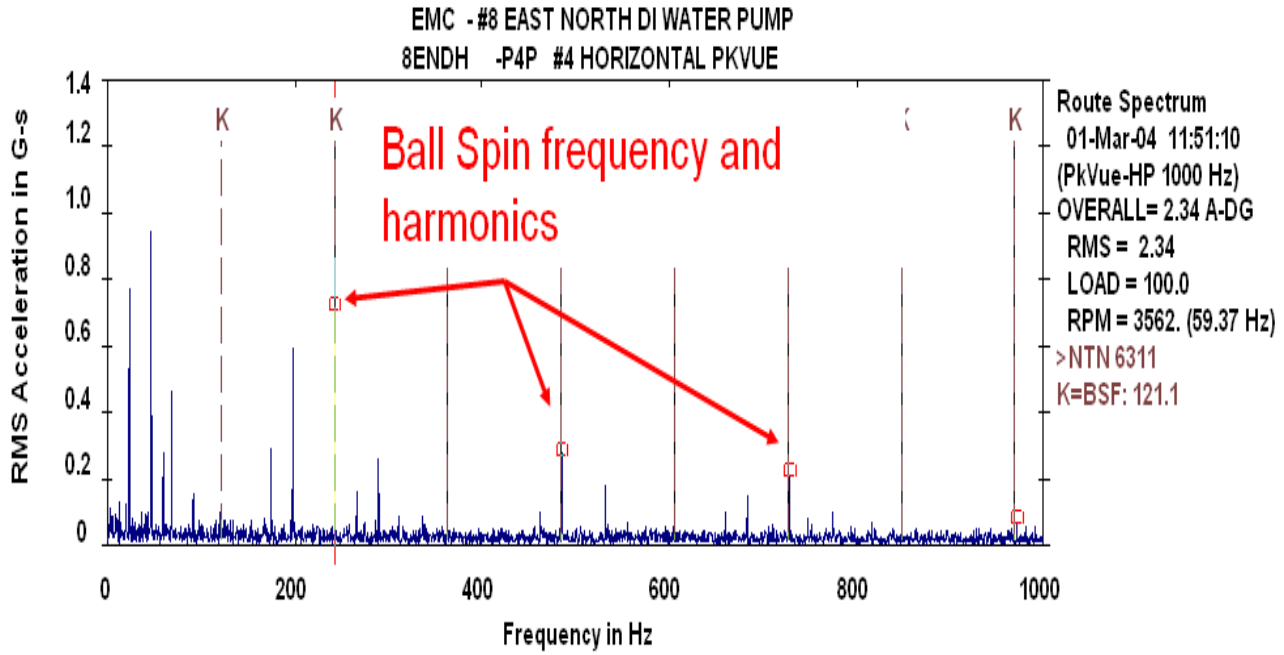
Roller Defect



When a roller develops a defect within an antifriction bearing, the defect will typically contact both races of the bearing as it makes a single revolution. Therefore, you will typically see peaks in the spectrum at 2 x the BSF, along with harmonics of BSF.



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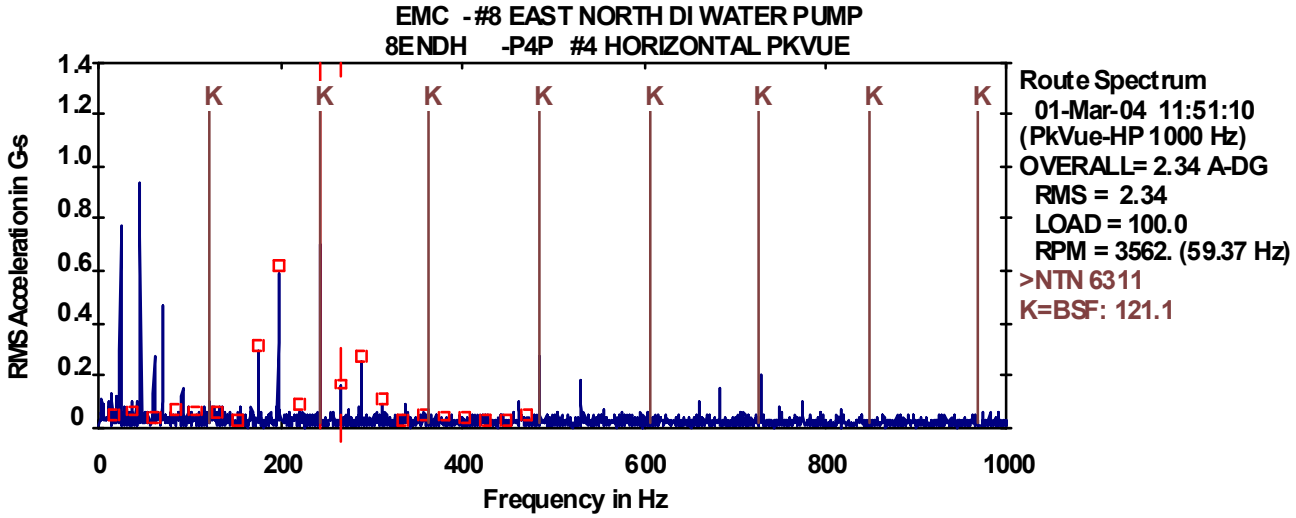
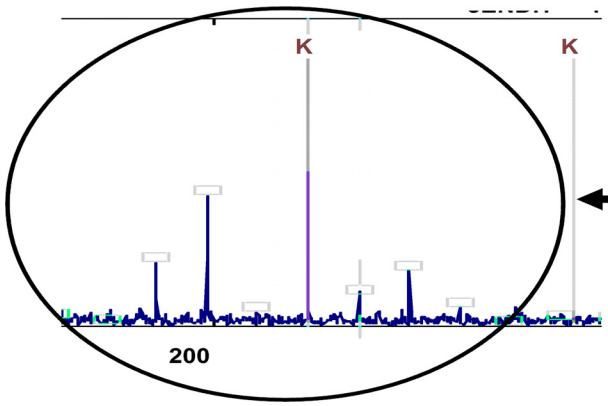
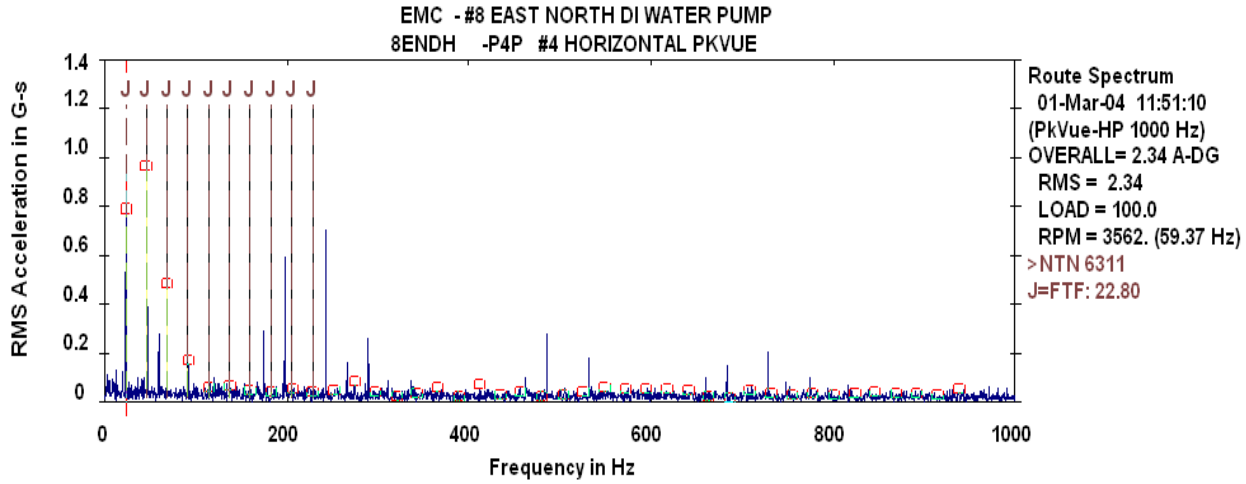




Cage frequency sidebands are present around the peak at 2x BSF.



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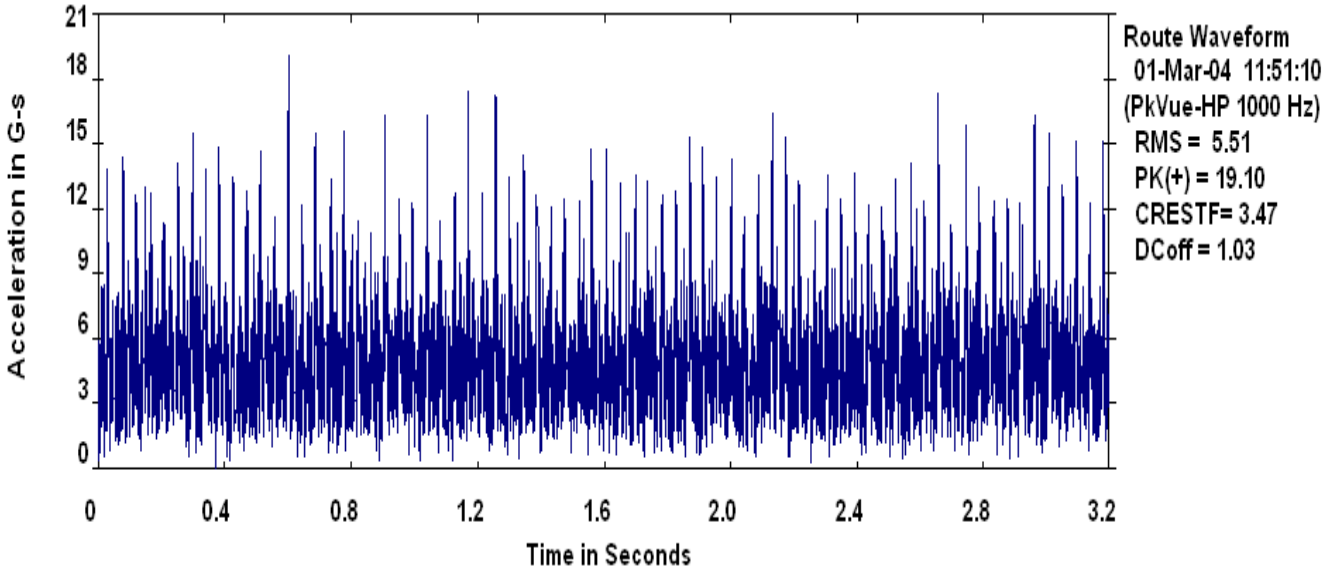
As a rolling element goes in and out of the load zone of the bearing, the force with which the defect contacts the races will increase and decrease. This causes amplitude modulation at the fundamental cage frequency of the bearing. In this case, as with most cases, FTF peaks are in the spectrum not because of any problem with the cage, but as a modulation frequency of the roller defect.

It is very rare to have a roller defect in the PeakVue spectrum without the presence of either peaks at the calculated FTF and harmonics or sidebands around the BSF at the FTF. This is one reason that it is important to have at least 15 revolutions of the shaft in the waveform.

In order to be able to identify a period in the waveform as being truly periodic, it needs to occur five times within that waveform. Since the typical FTF is somewhere between .4 and .45 times the shaft turning speed, 15 revolutions of the shaft would ensure that there are at least five periods of the Fundamental Train within the waveform.



The shaft RPM is 3562. The Peak value in the waveform is 19 G's. Meaning this defect has reached well over the fault stage for this type of defect and should be replaced at the next scheduled down time.

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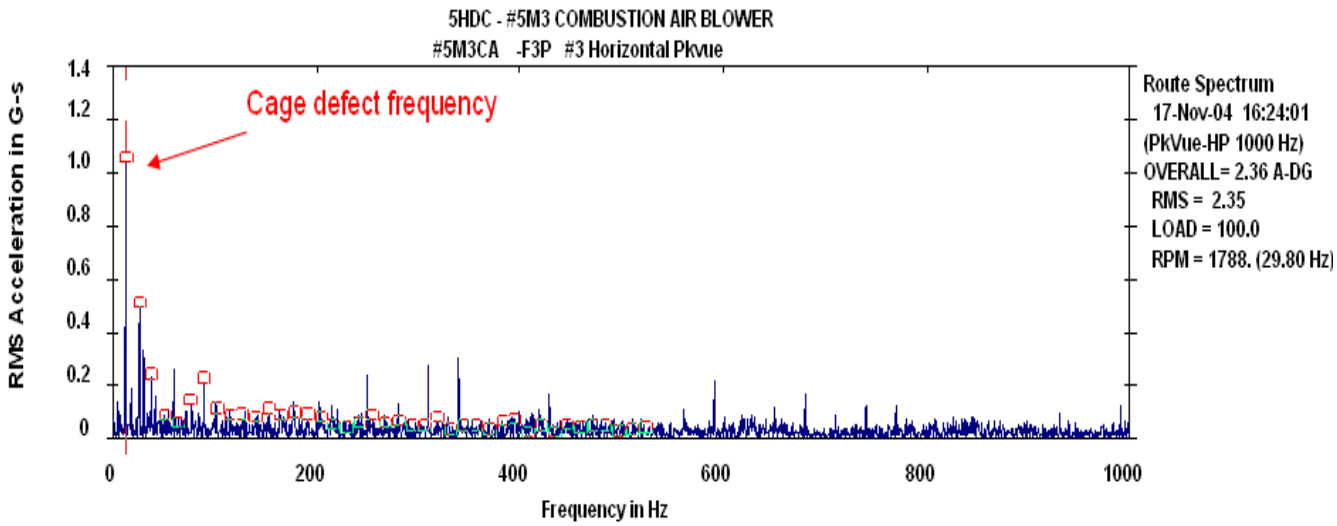
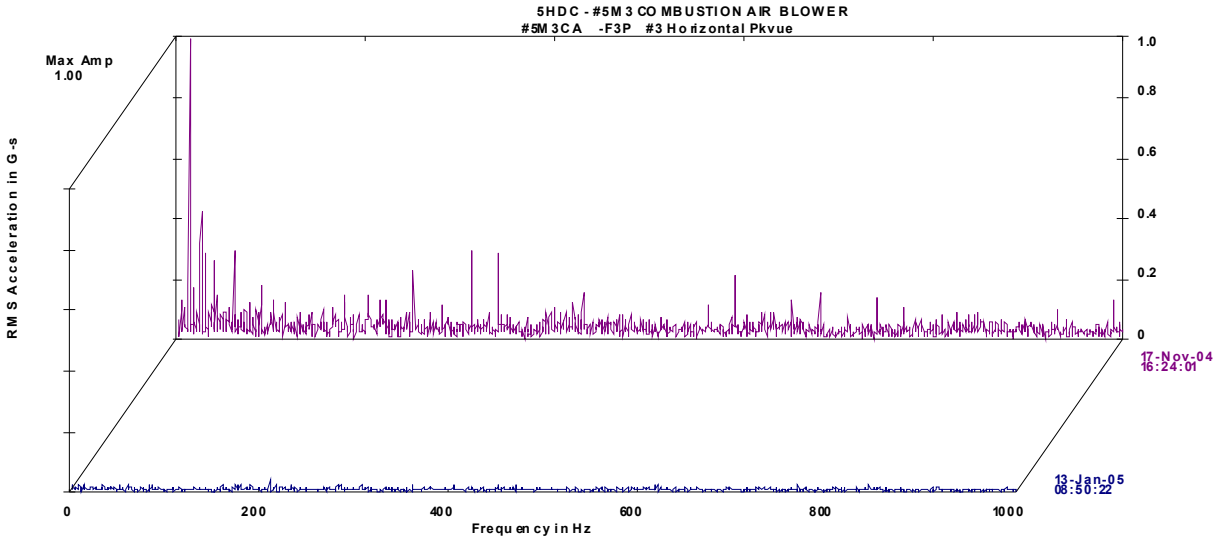
Cocked Bearing



The presence of a peak at the FTF without any other significant peaks would usually indicate some kind of bent shaft, misalignment, soft foot, pipe strain, or anything that would put axial load on a bearing that was not designed for axial loading. This data was taken on a brand new installation. A physical inspection made with a dial indicator revealed that the bearing was cocked on the shaft by 12 mils.



After the bearing was replaced, the unit was checked again. The comparison of the new data to the original data is convincing.

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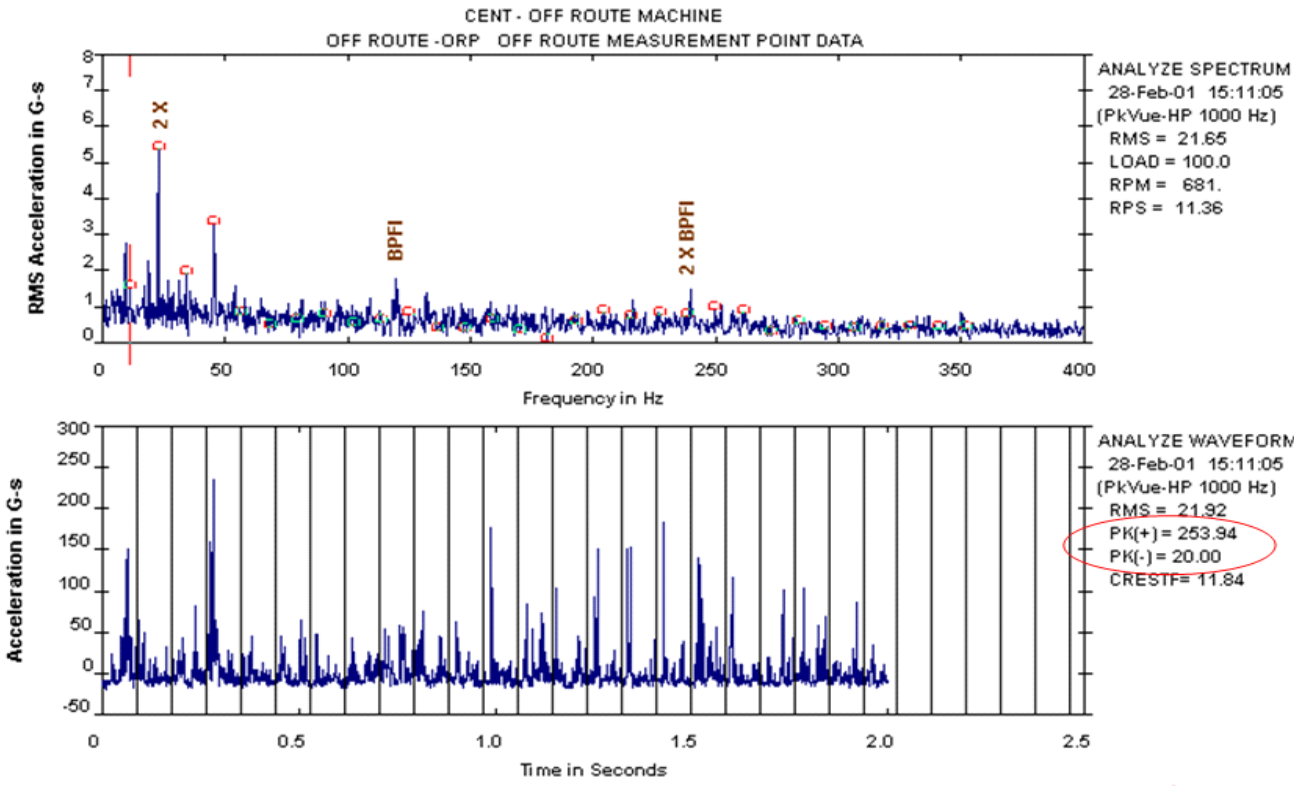
Lubrication Defect

Lubrication defects are characterized in PeakVue data by the presence of very high waveform amplitudes with very little periodic content. The presence of peaks in this PeakVue spectrum is attributed to the fact that this is a split-type bearing. Notice the amplitude in this waveform is over 250 G's.



The properly lubricated bearing situated right next to the previous bearing has an amplitude of only 2 G's. The viscosity of the oil in the previous bearing was found to be too low for the application.

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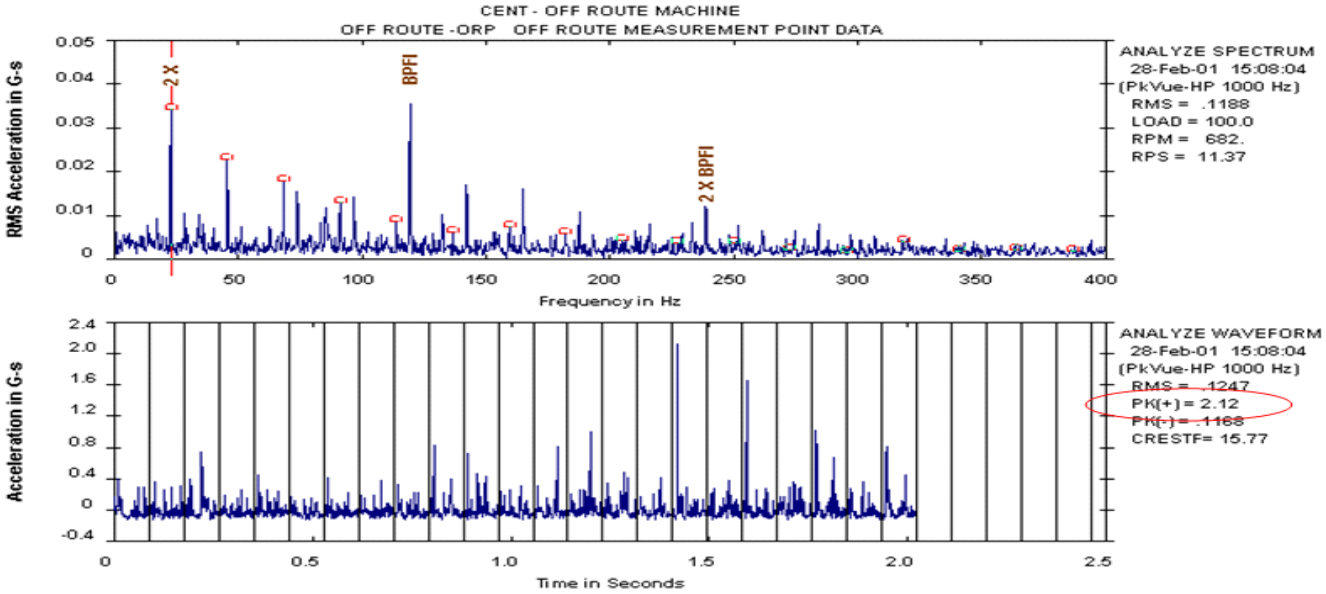
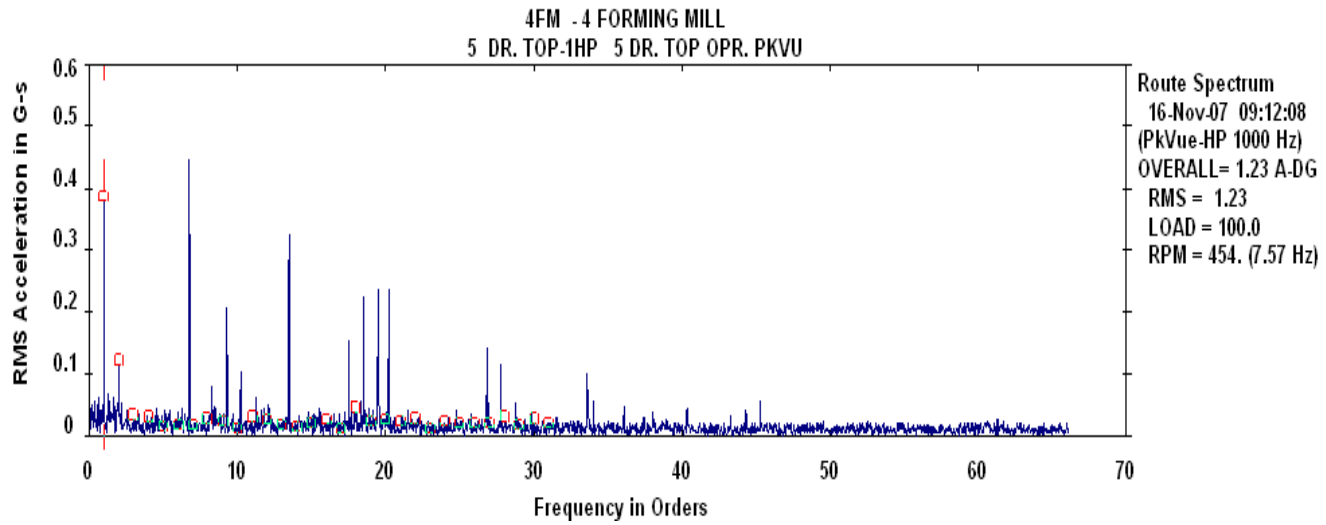
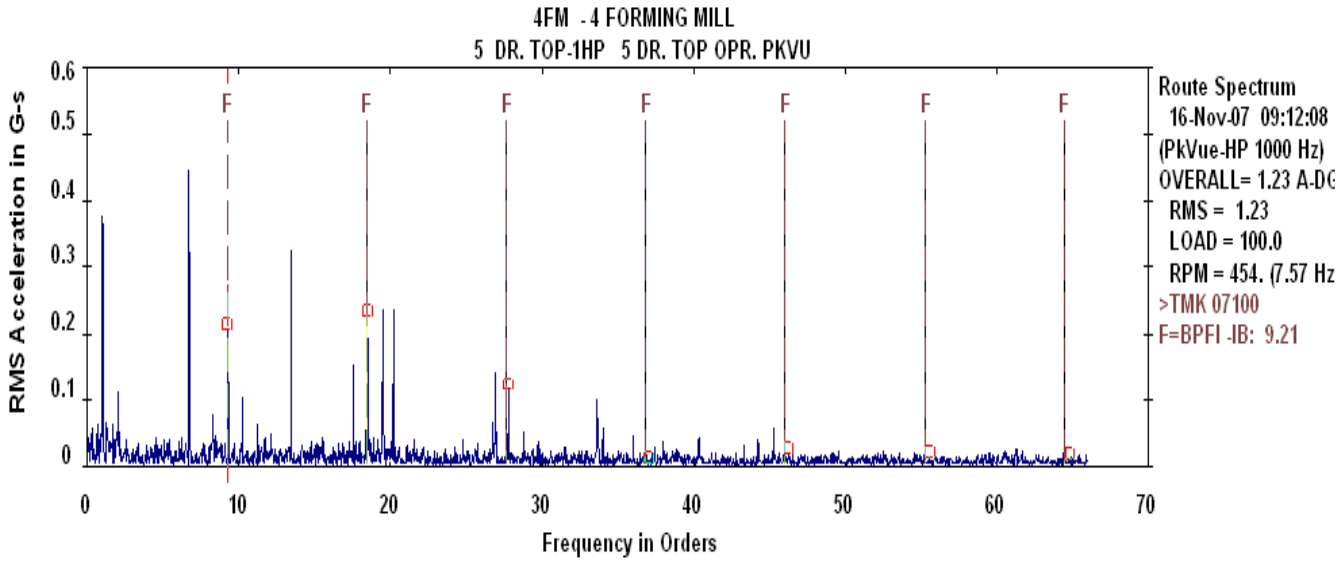

CENT - OFF ROUTE MACHINE
OFF ROUTE -ORP OFF ROUTE MEASUREMENT POINT DATA
RMS Acceleration in G-s
2 X
BPFI
2 X BPFI
ANALYZE SPECTRUM
28-Feb-01 15:08:04
(PkVue-HP 1000 Hz)
RMS = .1188
LOAD = 100.0
RPM = 682.
RPS = 11.37
Frequency in Hz
Acceleration in G-s
ANALYZE WAVEFORM
28-Feb-01 15:08:04
(PkVue-HP 1000 Hz)
RMS = .1247
PK(+) = 2.12
PK(-) = .1168
CRESTF= 15.77
Time in Seconds

Electrical Fluting



Electrical fluting is visually characterized by distinct ridges on one or both of the races caused by arcing inside the bearing. In this case, there were multiple faults. There are peaks at 1 x turning speed and harmonics which are probably related to the inner race faults.



Here the peaks at the BPFI and harmonics are highlighted along with turning speed sidebands around the BPFI harmonics.

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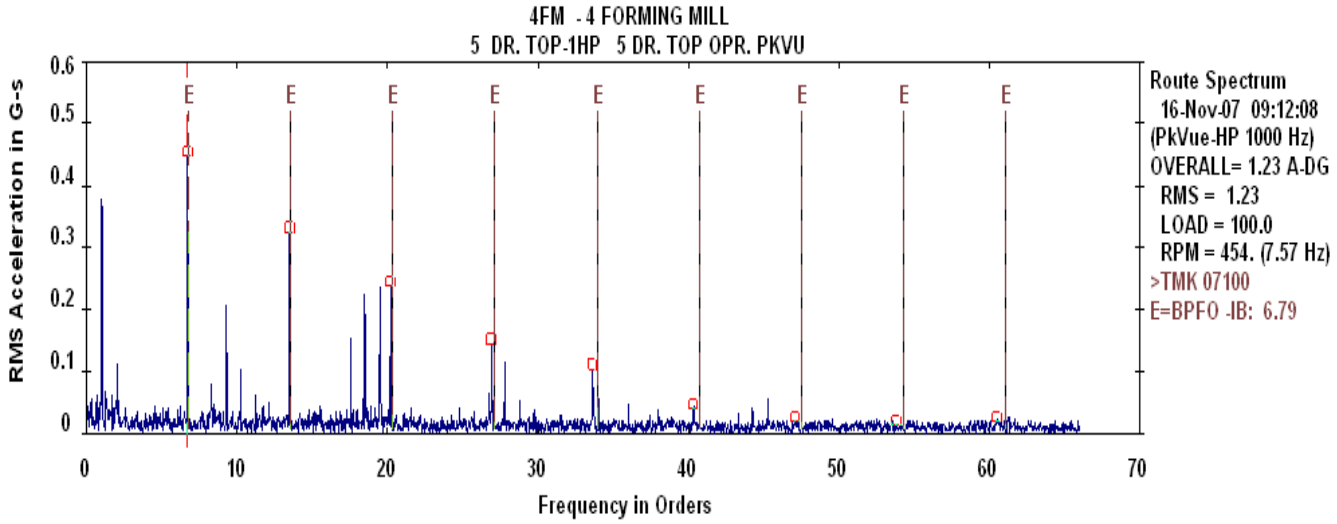
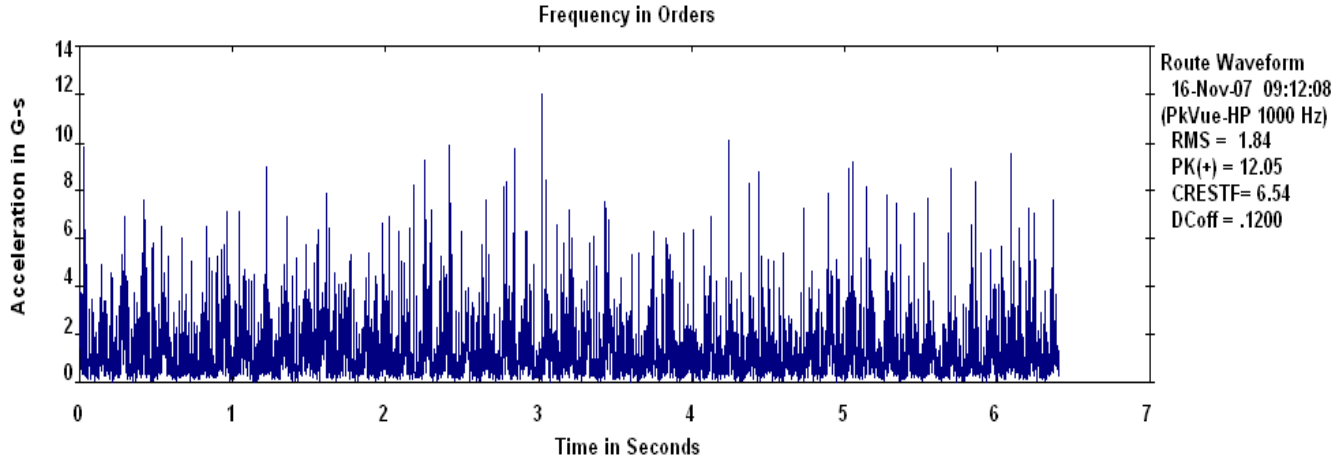


Outer race defects are present as well.



The corresponding PeakVue Max Peak waveform amplitude is 12 G's. The shaft turning speed is 454 RPM. The calculated alert level should be about 1.4 G's and the fault level 2.8 G's.

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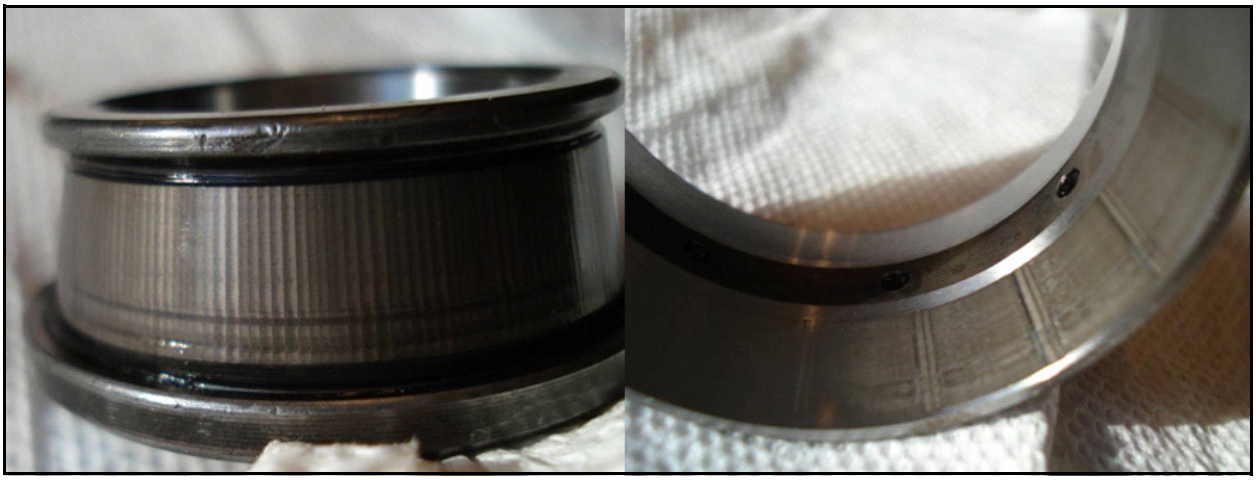
In this case, both outer race and inner race defects are clearly visible.



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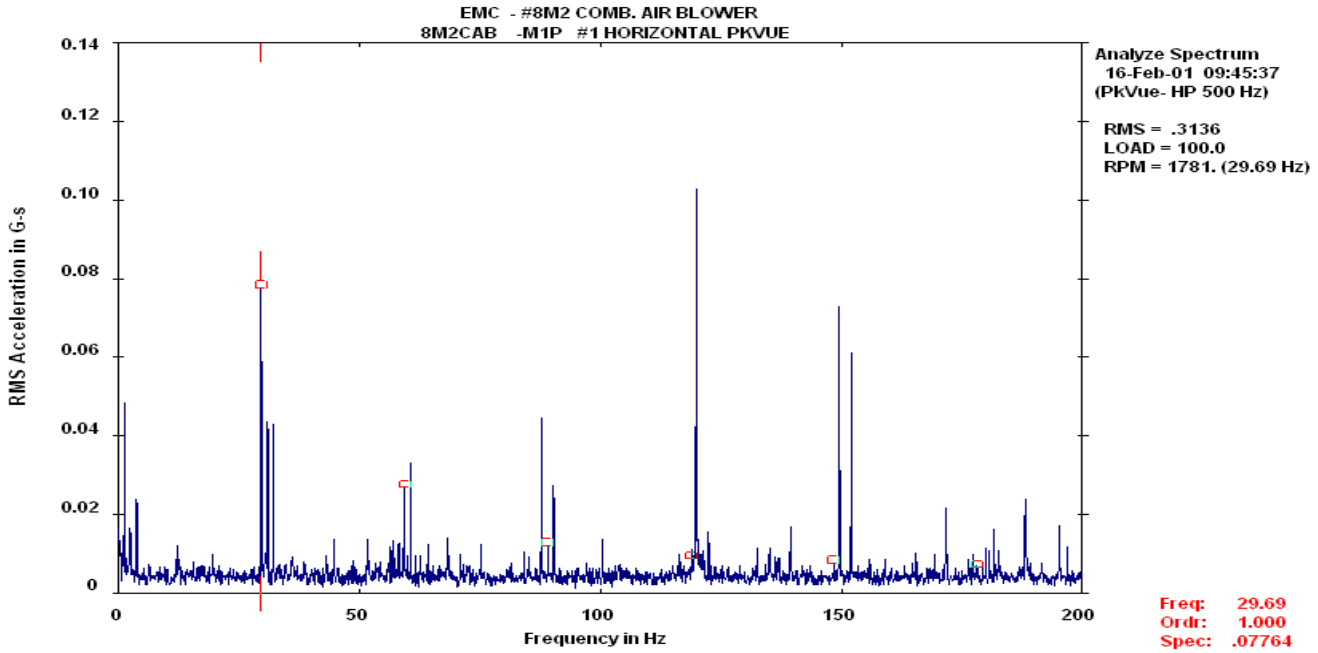
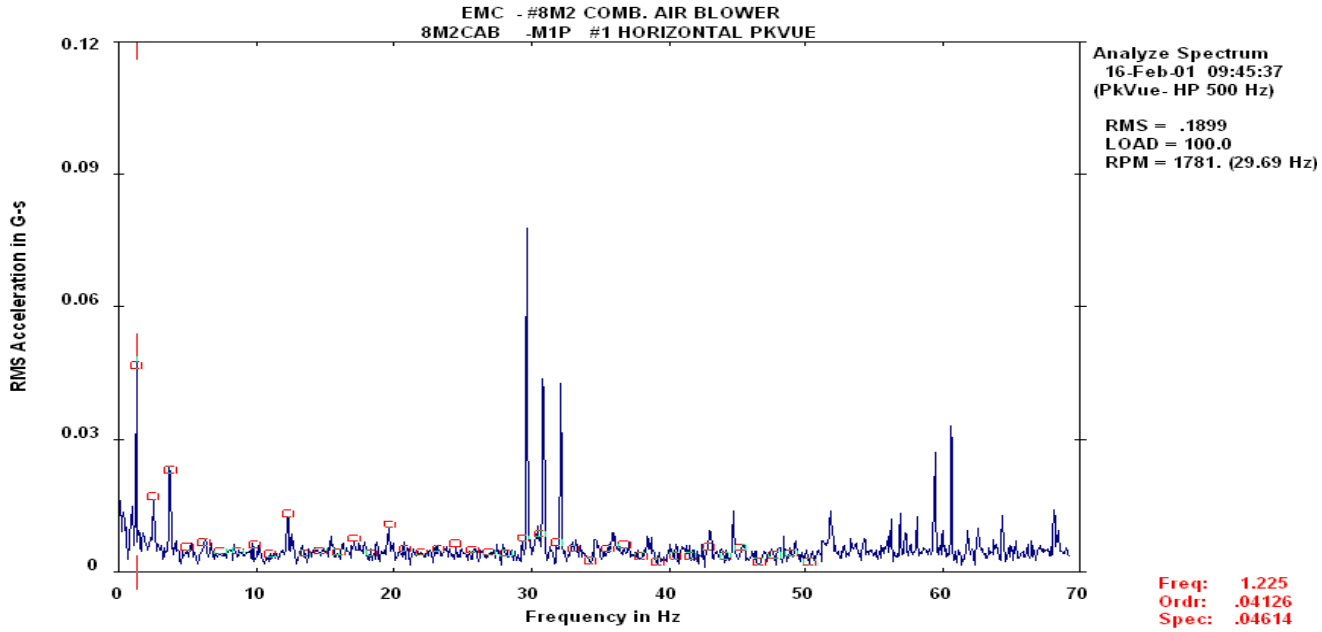
Broken Rotor Bars



There is a peak at 1 x turning speed, but of more interest are the very low frequency sub-synchronous peaks and the sidebands around the turning speed harmonics.



The first low frequency peak is at 1.225 Hz and the others are harmonics of that frequency.

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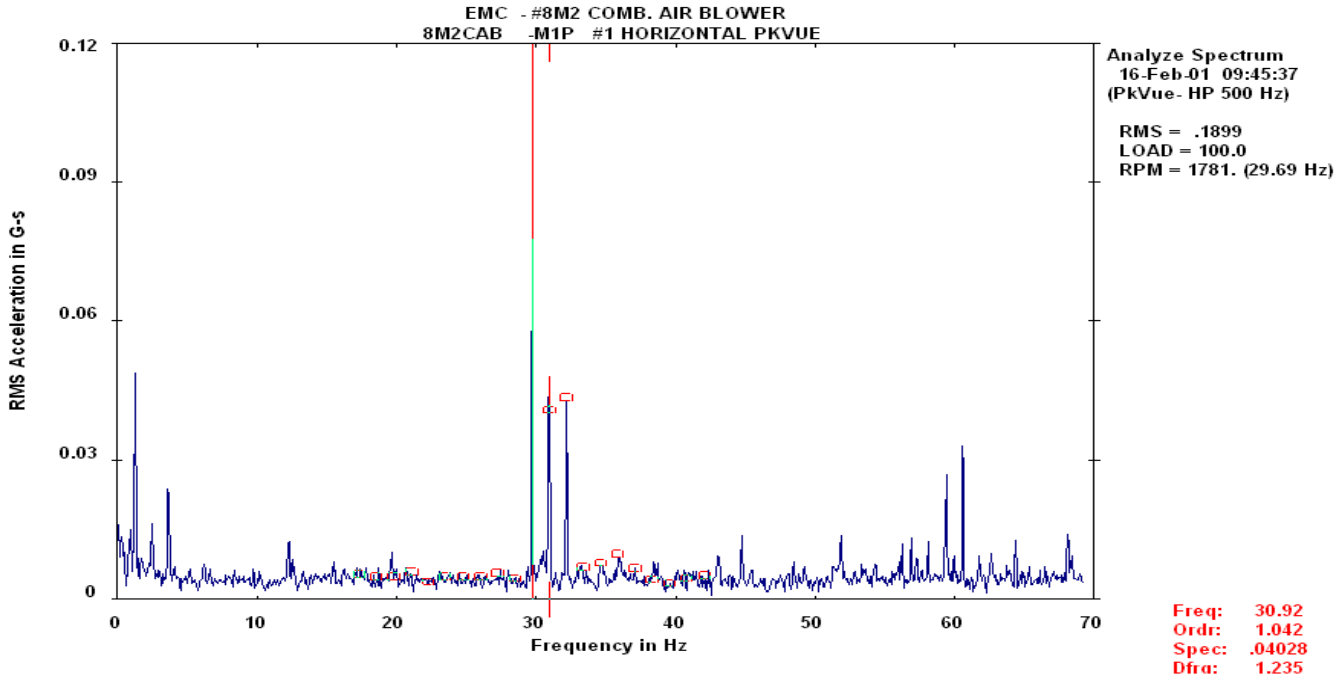

The sidebands close to the peak at 1 x turning speed are also spaced at 1.2 Hz. Based on this motor's turning speed of 1781 RPM, it has four poles and a slip frequency of .3 Hz. The frequency of 1.2 Hz is pole pass (number of poles x slip) frequency. Current tests confirmed that this motor did indeed have multiple broken rotor bars.

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Review Questions

1. What is the PeakVue sample rate?

2. True or False? Imbalance can be measured using PeakVue.

3. Which type of integration works best with PeakVue, Analog or Digital?

4. How many averages are usually acquired for PeakVue?

5. Which type of filter is typically used with PeakVue?

6. Which amplitude unit is used to analyze PeakVue data on very low RPM equipment?

7. How many shaft revolutions will be stored in a waveform that is acquired using the following spectral parameters? RPM = 1780; Fmax = 1000 Hz; LOR = 1600

8. Which type of magnet is preferred for PeakVue data acquisition?

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Antifriction Bearings

Section 8

Objectives

- Calculate rolling element bearing defect frequencies.
- Recognize bearing defect frequencies in spectral data.
- Discuss factors related to bearing life.
- Determine defect severity based on spectral frequency patterns and waveform amplitude.

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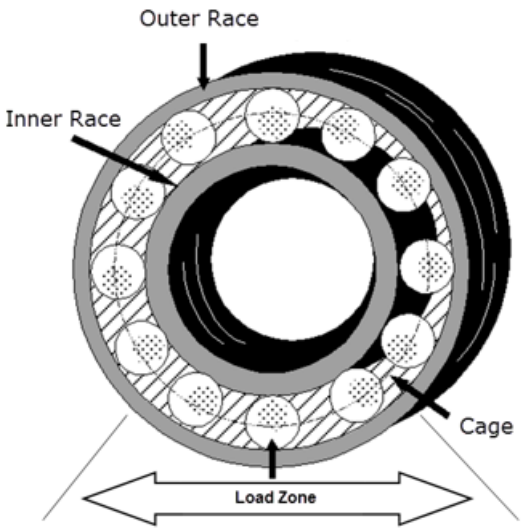
Bearing Components and Defect Frequencies



Rolling element bearings have specific bearing failure modes that can be observed in the spectral and the waveform data. Bearing defect frequencies are different than some other vibration frequencies present in a frequency spectrum because bearing frequencies are truly defect frequencies. These frequencies will not appear unless the bearings are actually defective. If the bearing is in good condition, has been installed properly and is properly lubricated, these fault frequencies should not appear in the vibration spectrum. Each defect frequency relates to a particular component of the bearing. For instance, if the BPFO defect frequency appears in the spectrum, there is a defect on the outer race. If BSF shows up in the spectrum, there is a defect on one or more of the rollers. This holds true for all of the defect frequencies except for the FTF, which can show up in the frequency spectrum for reasons other than a defect on the cage itself.

FTF	Fundamental Train Frequency or Cage Frequency
BSF	Ball Spin Frequency
BPFO	Ball Pass Frequency Outer Race
BPFI	Ball Pass Frequency Inner Race

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The calculation of these fault frequencies can be done manually using the following values. First, for a bearing with a rotating inner race and fixed outer race:

$$BPFI = \frac{Nb}{2} \cdot S \cdot (1 + \frac{Bd}{Pd} \cdot \cos\theta)$$

$$BPFO = \frac{Nb}{2} \cdot S \cdot (1 - \frac{Bd}{Pd} \cdot \cos\theta)$$

$$FTF = \frac{S}{2} \cdot (1 - \frac{Bd}{Pd} \cdot \cos\theta)$$

$$BSF = \frac{Pd}{2Bd} \cdot S \cdot [1 - (\frac{Bd}{Pd})^2 \cdot (\cos\theta)^2]$$

If the bearing has a stationary inner race and rotating outer race, the fault frequencies are the same. The exception is the Fundamental Train Frequency, which is listed below.

$$FTF = \frac{S}{2} \cdot (1 + \frac{Bd}{Pd} \cdot \cos\theta)$$

S = Speed, revolutions per second

Bd = Ball or roller diameter

Nb = Number of balls or rollers

Pd = Pitch diameter

Θ = Contact angle

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In most cases, the bearing fault frequencies can be found by searching the bearing database within AMS Machinery Manager using the Frequency Calculation tool.

BEARINGS in CSI Warehouse: \...\SysData\CSI_CMP.WH

BRG ID	BEARING TYPE	#B/R	FTF	BSF	BPFO	BPFI
9441	DGE 066220	22	.443	4.191	9.740	12.260
3214	FAG 6220	10	.409	2.664	4.093	5.907
81271	FAG 6220	10	.409	2.665	4.093	5.907
81272	FAG 62200-2RSR	7	.361	1.661	2.528	4.472
81273	FAG 62201-2RSR	7	.364	1.697	2.545	4.455
81274	FAG 62202-2RSR	8	.381	1.990	3.051	4.949
81275	FAG 6220-2RSR	10	.409	2.665	4.093	5.907
81276	FAG 6220-2Z	10	.409	2.665	4.093	5.907
81277	FAG 62203-2RSR	8	.382	1.994	3.053	4.947
81278	FAG 62204-2RSR	8	.382	1.992	3.052	4.948
81279	FAG 62205-2RSR	9	.397	2.315	3.570	5.430
81280	FAG 62206-2RSR	9	.396	2.311	3.568	5.432

Important Relationships

1. Bearing defect frequencies are almost always non-synchronous. (Except for the FTF which is sub-synchronous.)

2. The sum of the BPFO and BPFI is equal to the number of rolling elements.

BPFO + BPFI = # balls or rollers

3. For the Inner Race rotating, the FTF is equal to the BPFO divided by the number of rolling elements.

FTF = BPFO ÷ # balls or rollers

4. For the Outer Race rotating, the FTF is equal to the BPFI divided by the number of rolling elements.

FTF = BPFI ÷ # balls or rollers

5. The BPFI is always the highest of the four bearing defect frequencies.

In addition to these defects, vibration analysis can help in the detection of excessive internal clearance, improper lubrication and bearing misalignment. Identification of these faults in the vibration spectrum is not easy. The particular fault may not show up at its calculated frequency, especially when the defect is very minor. There may not be enough vibration energy to generate a spectral peak at that frequency.

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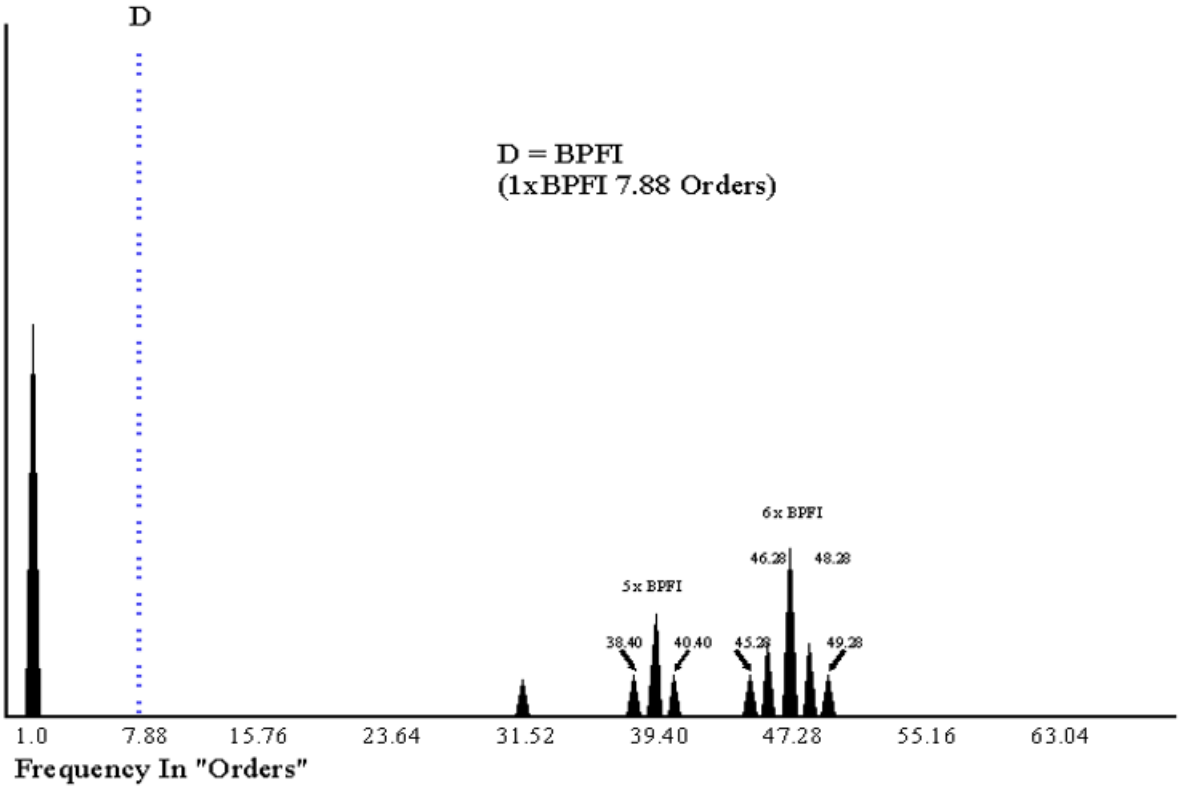
Typical Failure Progression of Antifriction Bearings

Traditional spectrum analysis using pattern recognition of velocity spectra.

Mode 1



In the first mode, the first signs of a defective bearing will appear in the spectral data, typically at **higher frequency multiples** of the fundamental bearing defect frequency. Often, there is simply not enough energy at the defect frequency to generate a peak at that frequency in the spectrum. However, because the defect causes periodic impacting, a family of spectral harmonics is generated. The harmonics that are closest to the bearing's natural frequency become amplified. It is these amplified harmonics which are usually seen first in the vibration spectrum. The vibration analyst should always pay close attention whenever a non-synchronous harmonic family appears in the spectrum. In the first mode, the defect is usually very minor. A visual inspection of the race or roller surface may not be able to identify the defect at all.

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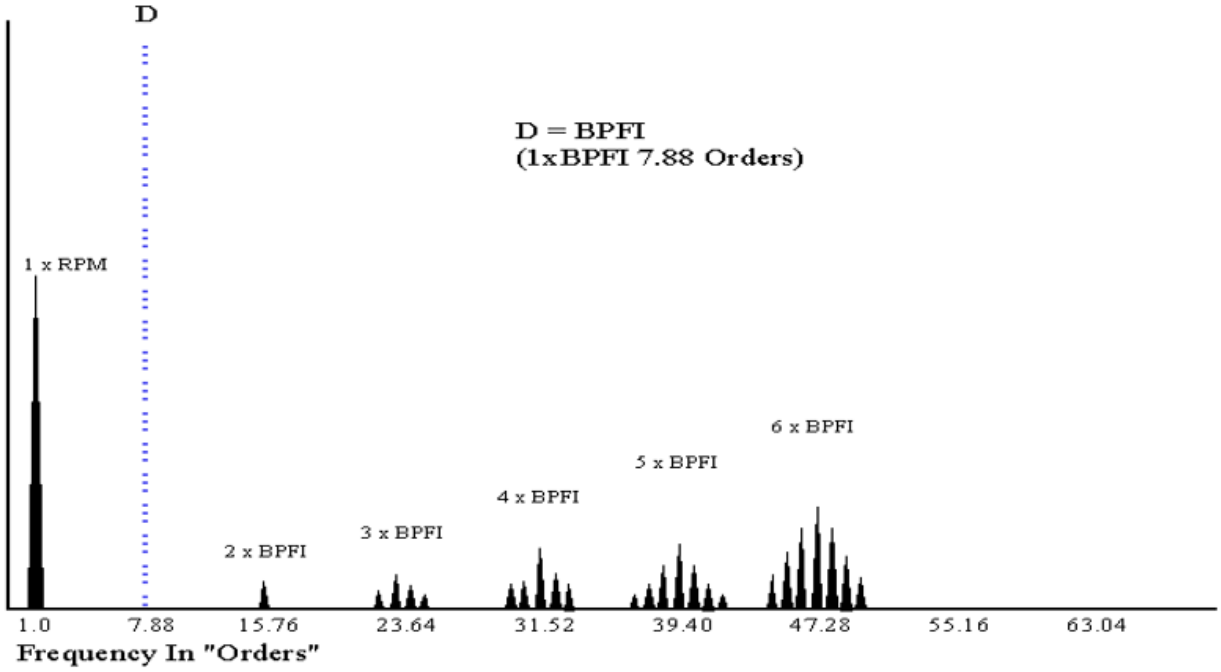
Mode 2



In the second mode, more harmonics of the **defect frequency** appear in the spectral data. With certain defects, notably BPFI and BSF, sidebands will appear around the defect frequency harmonics. These sidebands are generated due to the modulation caused by the defect going in and out of the load zone of the bearing. An inner race defect will typically modulate at 1 x RPM of the shaft, because a defect on the inner race will travel in and out of the load zone once per revolution of the shaft. A roller defect (BSF) will modulate at the FTF, because a single roller will travel in and out of the load zone once per revolution of the train or cage. Sometimes the sideband amplitudes will be higher than the harmonics themselves. This is usually indicative of a more severe condition. Also, as the bearings degrade, the actual defect frequencies may not exactly match the pre-calculated defect frequencies. This is because degradation of the bearing has caused the internal bearing geometry to deviate from the original design.

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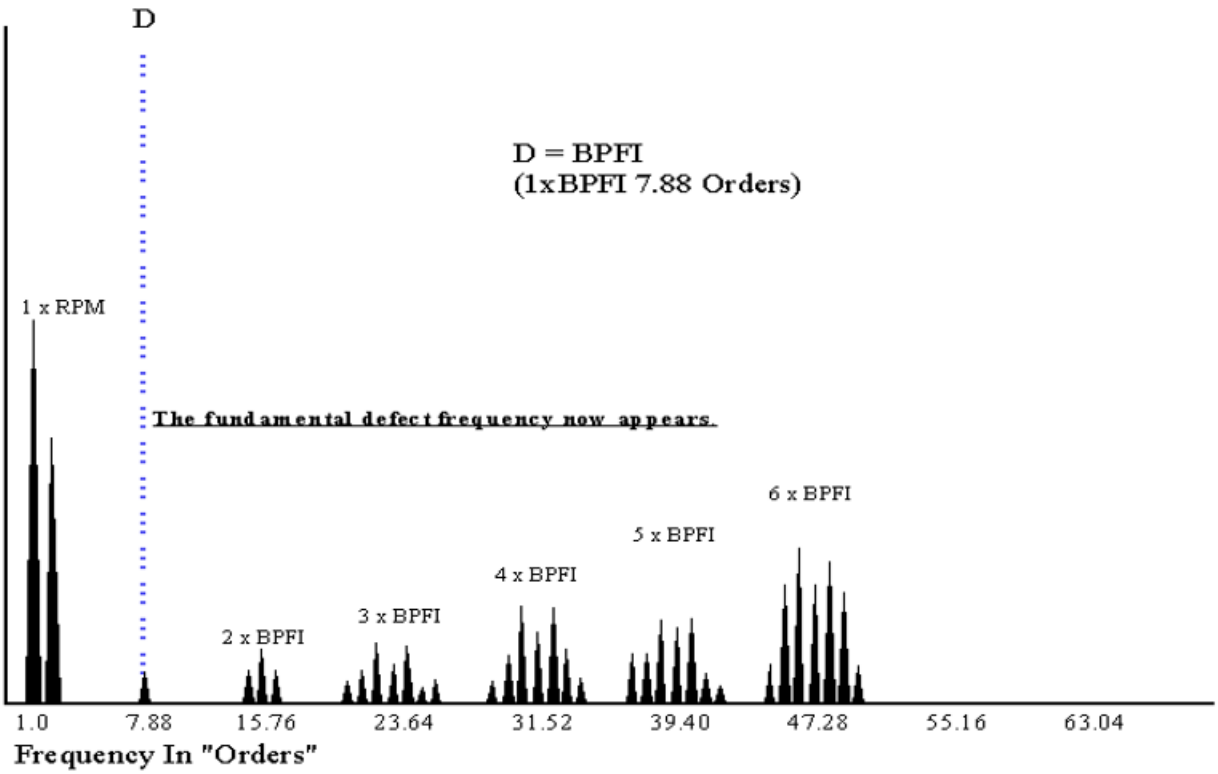
Mode 3



Mode 3 can be identified by the appearance of the **fundamental bearing defect frequency** in the spectral data, in addition to the defect harmonics and sideband frequencies. This is the mode at which corrective action should be scheduled in order to prevent further damage, unscheduled outage and/ or possible catastrophic failure. In most cases, the bearing is not bad yet and may not exhibit any signs of failure. At this stage, the bearing has become unpredictable, and since the remaining bearing life cannot dependably be predicted, it is best to replace the bearing. If bearing replacement during outage is not possible, the bearing should be monitored closely for signs of failure until an outage can be scheduled.

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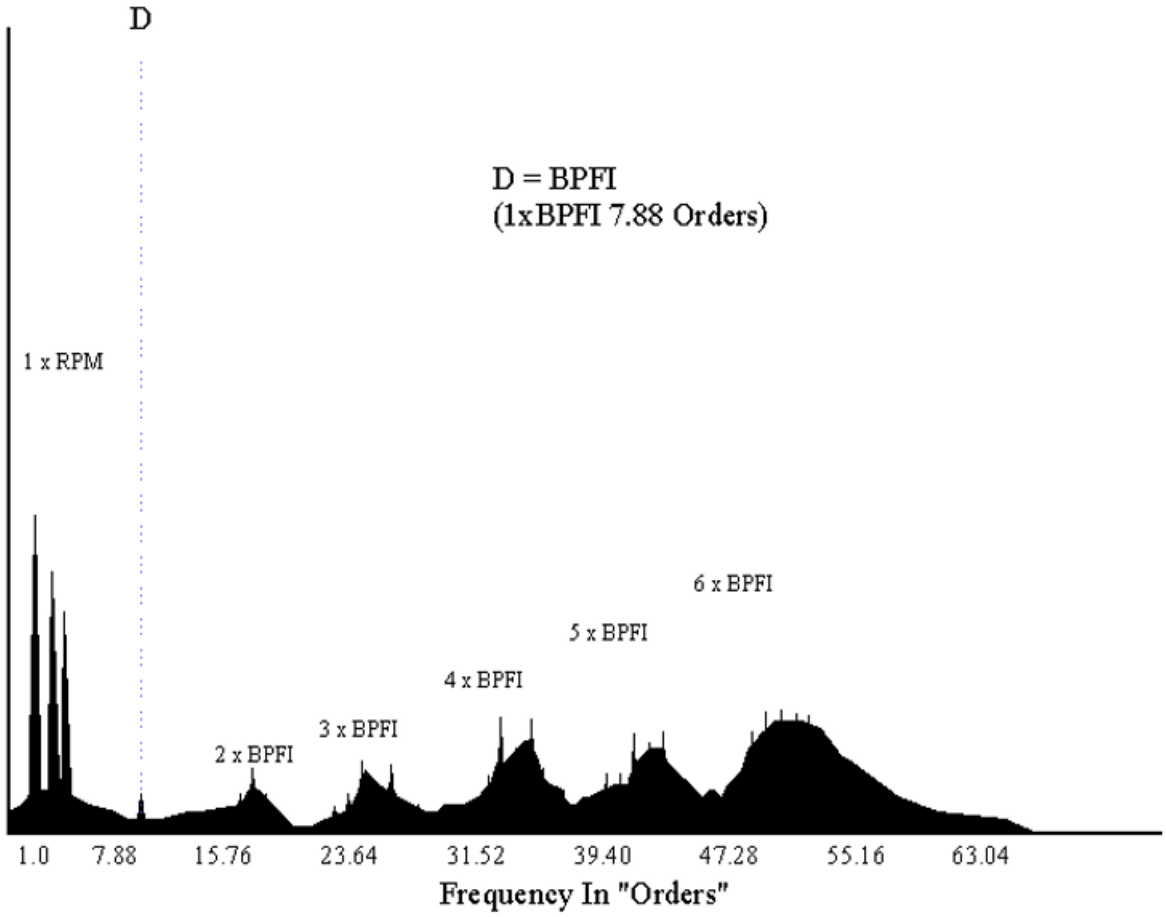
Mode 4



As the bearing continues to degrade, the **internal clearances** may begin to increase, which accelerates the deterioration of the bearing components and changes the frequency of the defects. This may cause the fault frequency markers to not line up. The peaks in the spectrum may seem to decrease in amplitude and become less distinct as the noise floor increases. This increased broadband vibration energy represents a much more random vibration signal. In many cases, the trend of the overall vibration energy will decrease at this stage, and the bearing may seem to be getting better. The vibration analyst needs to pay close attention at this stage, since failure may occur at any time. Bearing replacement should be completed as soon as possible. There is a very good chance that shaft and housing damage are present as well. To prevent this additional damage, bearings should be replaced at stage three whenever possible, to maintain equipment reliability and predictability.

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It is important to understand that each bearing may progress through the failure modes at different rates.

When a bearing defect is detected, data should be collected more often, so that all of the stages are able to be identified. Every bearing failure is somewhat different. Often, an analyst fails to capture some of the failure stages due to their occurrence between data collection intervals or changes happen very quickly.

Other Failure Modes

There are other modes of failure (before mode 1 and after mode 4) that may occur.

- The vHFD trend data may increase even **before mode 1**. Prudent analysts will utilize this tool effectively.
- **Before mode 1**, properly collected PeakVue data will show indications of bearing problems.
- In **mode 5**, broadband energy can disappear, and harmonics of 1 x turning speed may appear.
- **In mode 6**, harmonics of turning speed may disappear, broadband energy down towards 1 x TS may appear.
- **In mode 7,** even the broadband energy disappears and may leave only a small peak at 1 x TS along with a flat line. (No need to fix this bearing, it has fixed itself.)

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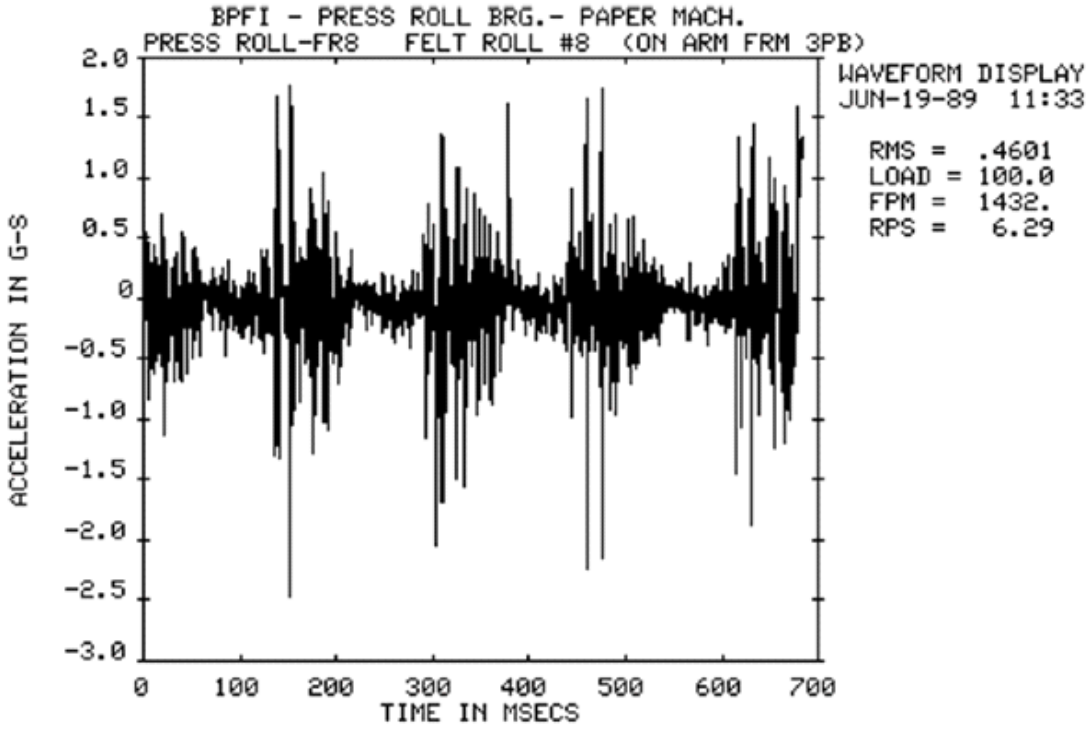
Determining Defect Severity

Along with spectral pattern recognition, an analysis of the waveform data can be very helpful when trying to assess the severity of any antifriction bearing defect.



This waveform shows the classic impact and ring down pattern often seen due to the rolling element impacting the defect site. The waveform should be acquired and analyzed in acceleration units and evaluated for amplitude to determine the severity of the defect. When determining the impact amplitude, the Peak to Peak G swing is usually identified. In this example, the positive peak value is about 1.75 g's, and the negative peak value is about -2.5 g's. This means the Peak to Peak acceleration value is about 4.25 g's. This Peak to Peak waveform value can be trended using AMS Machinery Manager and is generally seen as the most effective trend parameter available for trending bearing defect severity. However, it should be understood that acceptable waveform amplitudes are different for different machines based on a number of factors.

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Understanding Amplitude Levels

Bearing Types

It is helpful to know what **bearing types** are in the machines in order to more accurately identify the bearing fault frequencies displayed in the data. Once the defects are identified, it is not always possible to only trend the amplitude change over time. Some bearings may fail when the waveform amplitude is less than 2 g's. Others may run for extended periods of time with waveform amplitudes in excess of 20 g's.

Amplitude Variations

One reason for **amplitude variations** is the vibration energy transmission path. Amplitudes for the same defect severity will be lower if the vibration has traveled through several feet of the machine than if the sensor were mounted just a few inches from the bearing. Also, inner race defects may appear at lower amplitudes than outer race defects, due to the different transmission path.

Mass, Stiffness and Damping

Another reason that amplitudes alone cannot be relied upon to provide accurate defect severity is the differences in each machine's mass, stiffness and damping properties. These are also known as the **system properties**. Even machines that are side by side, with the same basic design, could have slightly different system properties that could affect the measured amplitudes of similar sized bearing defects on each machine.

Sir Isaac Newton's Second Law

This law states that force is equal to its mass multiplied by the acceleration of the mass. The equation is written:

$$\mathbf{F = M \times A}$$

Where: **F** = Force

M = Mass

A = Acceleration

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Using basic algebra, this means that the acceleration is equal to the force divided by the mass.

$$\mathbf{A = F/ M}$$

Calculate the acceleration of two different machines. The first machine has ten times more mass than the second machine, but the force being applied to each machine is equal. In this exercise, the force applied to each machine is 10.

For the first machine, which has a mass of 10, the equation is below. The acceleration measured on the first machine would be one g.

$$\mathbf{A = 10/ 10}$$

The second machine has mass equal to only one. The measured acceleration on the second machine would be 10 g's.

$$\mathbf{A = 10/ 1}$$

In summary, smaller machines which have less mass, tend to have higher acceleration values than larger machines, which have more mass.

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Bearing Life

The expected life of a bearing can be calculated using the following equation:

$$H = (C/ L)^3 \times (16667/ RPM)$$

Where: **H** = Life in hours

C = Bearing capacity in pounds

L = Bearing load in pounds

Based on this equation, there are two operational variables which affect bearing life: **Load** and **RPM**. These two variables are not used equally in the equation. Notice that the load value, after being divided into the capacity, is cubed. The RPM is not cubed.

As a result of this equation, if the bearing's load is doubled, the bearings calculated life will be eight times less. If the RPM is doubled, the calculated life will be only two times less. A bearing's load has much more of an effect on its expected life than does its RPM. It is common for vibration analysts to assume that high RPM machines will experience rapid bearing defect progression as compared to lower RPM machines. Many low RPM machines are also very large and massive, so these machines are just as likely, if not even more likely, to progress to failure quickly. This also explains how imbalance, misalignment, and excessive belt tension can greatly reduce equipment life.

PeakVue Analysis

One of the advantages of using Emerson's patented PeakVue technology to identify and evaluate anti-friction bearing defects, is that the spectral pattern recognition is more obvious and straight forward. Even in the first mode of a defect, the spectrum usually contains a dominant peak at the calculated defect frequency. Also, because of the special filtering and signal processing, peaks from other forcing functions such as imbalance and misalignment do not appear in the spectrum, making defect recognition that much easier. PeakVue analysis of anti-friction bearing defects involves analysis of both the spectral and time waveform data. The spectrum is analyzed to identify the type of defect or defects present. The waveform is analyzed for evaluation of severity. The following examples illustrate the spectral and waveform patterns of actual bearing defects using PeakVue technology.

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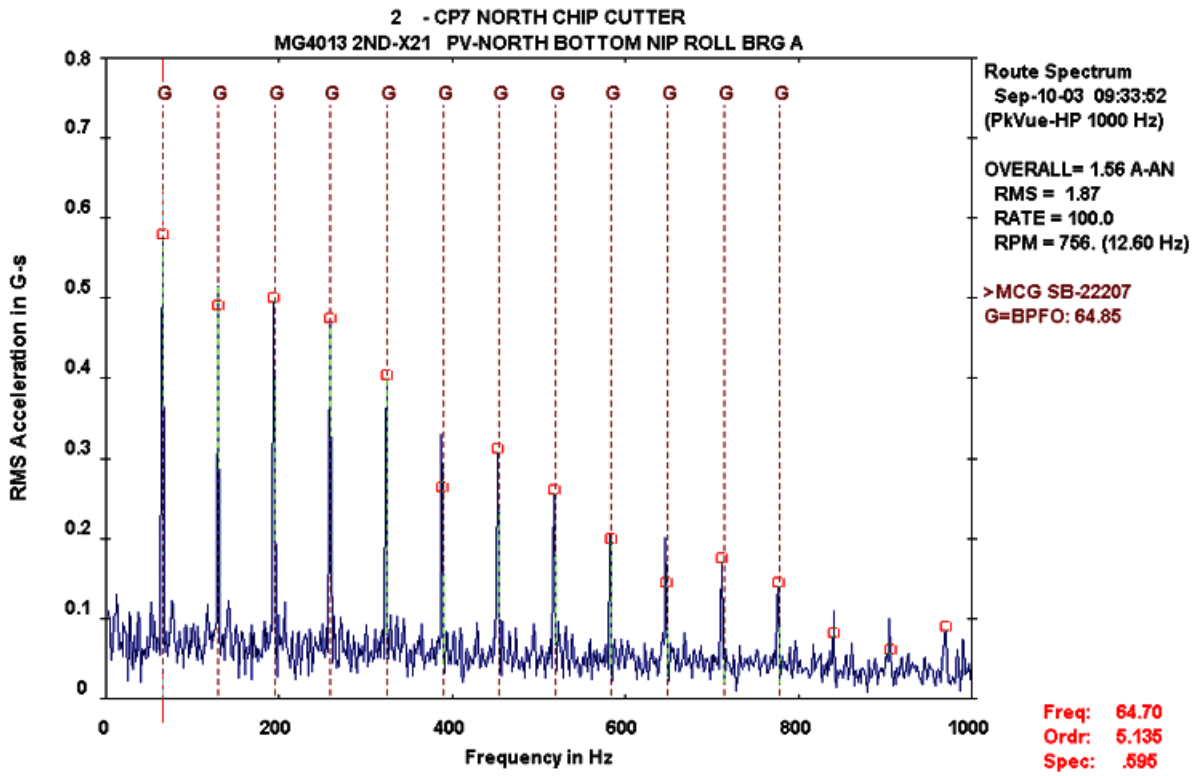
BPFO



When only an outer race defect is present, the spectrum should show a dominant peak at the bearing's calculated BPFO, usually accompanied by multiple harmonics of the BPFO. Note the absence of any other significant peaks in the spectrum. In most cases, a defect on the outer race will not cause significant modulation. The most common mistake that analysts make at this point is to assume the defect has reached a critical state because of the distinct spectral pattern. Remember, before any conclusion is made about the severity, the time waveform must be reviewed.

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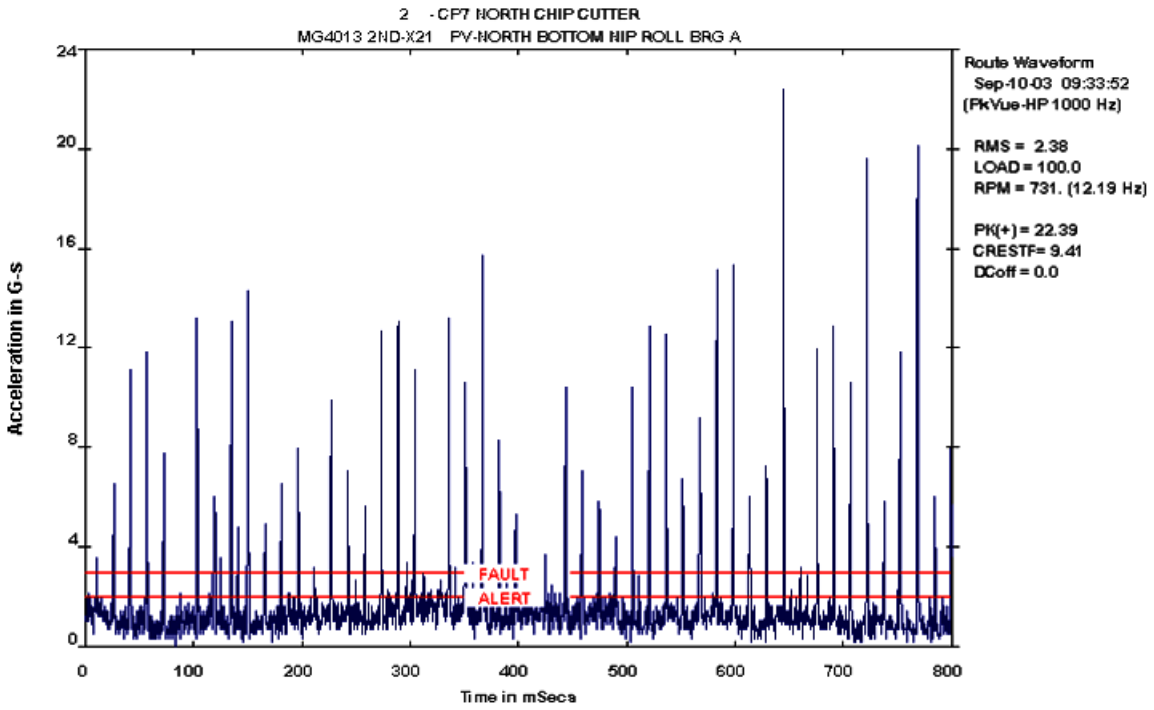

The time waveform Peak amplitude is 22.39 g's. At 731 RPM, this represents a severity about two times higher than the calculated fault level for this type of defect. A recommendation should be made at this time to replace the bearing at the machine's next scheduled outage. The trend data should also be evaluated to identify the rate of the severity progression. In many cases, the trend of the measurement point's Max Peak Waveform trend parameter is the most useful parameter to gauge the progression of an anti-friction bearing defect using PeakVue.

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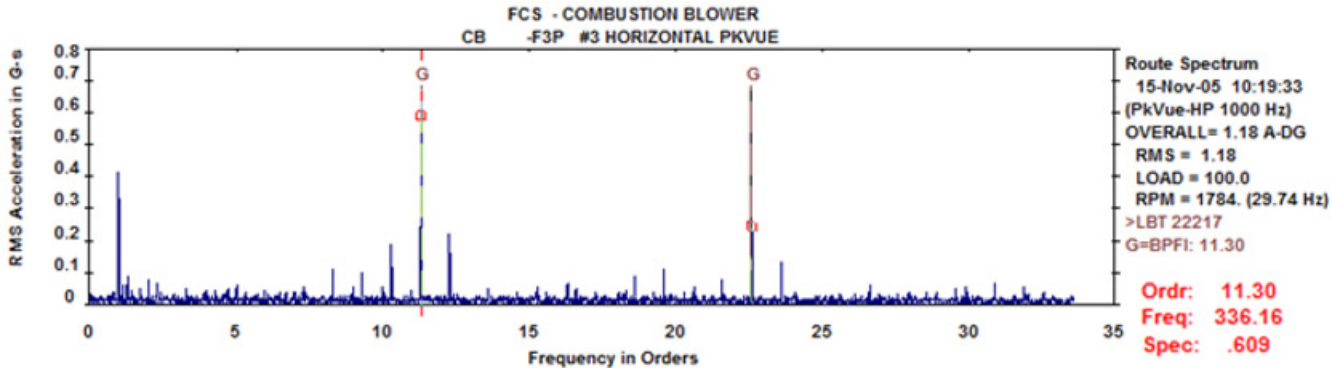
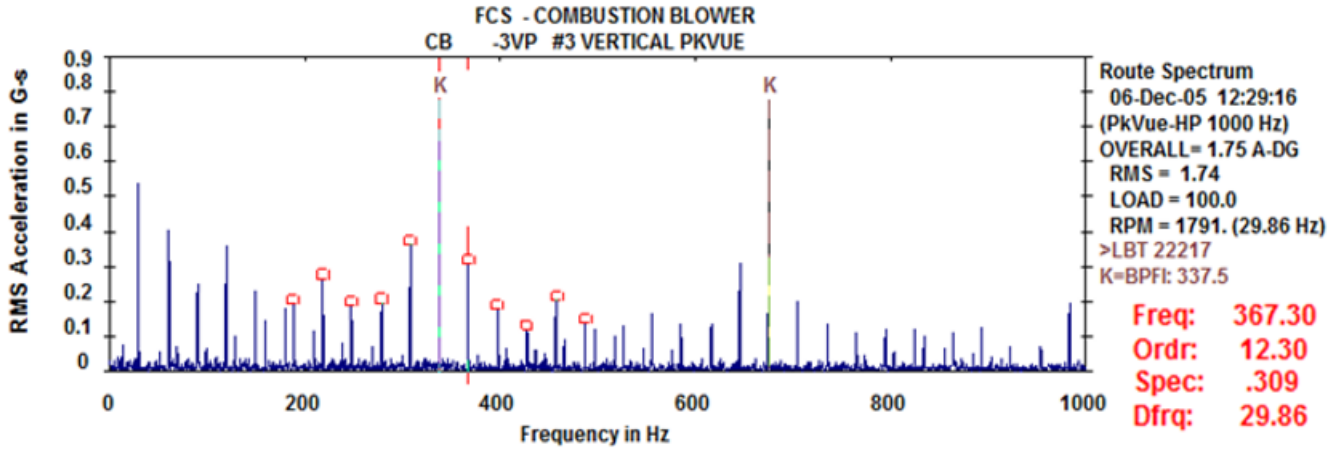
BPFI

When an inner race defect appears in the PeakVue spectrum, expect to see a dominant peak at the bearing's calculated BPFI. In this example, the BPFI is 11.3 orders. 1 x TS sidebands are present around the peak at the BPFI, which represent the once per revolution amplitude modulation that is typically associated with an inner race defect. In addition to the sidebands, there is also a discrete peak at 1 x RPM. This peak also represents the 1 x RPM modulation.



As the defect gets progressively more severe, more sidebands are present and multiple harmonics of turning speed also appear in the spectrum. Although this progression can easily be seen in the spectral data, the evaluation of severity is still best accomplished by an analysis of the waveform amplitude.



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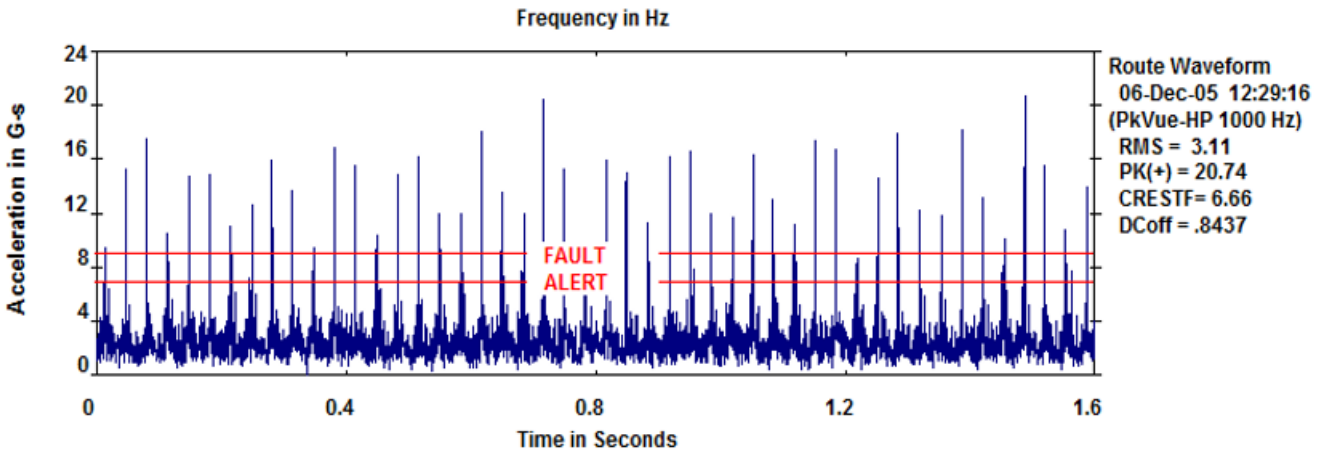
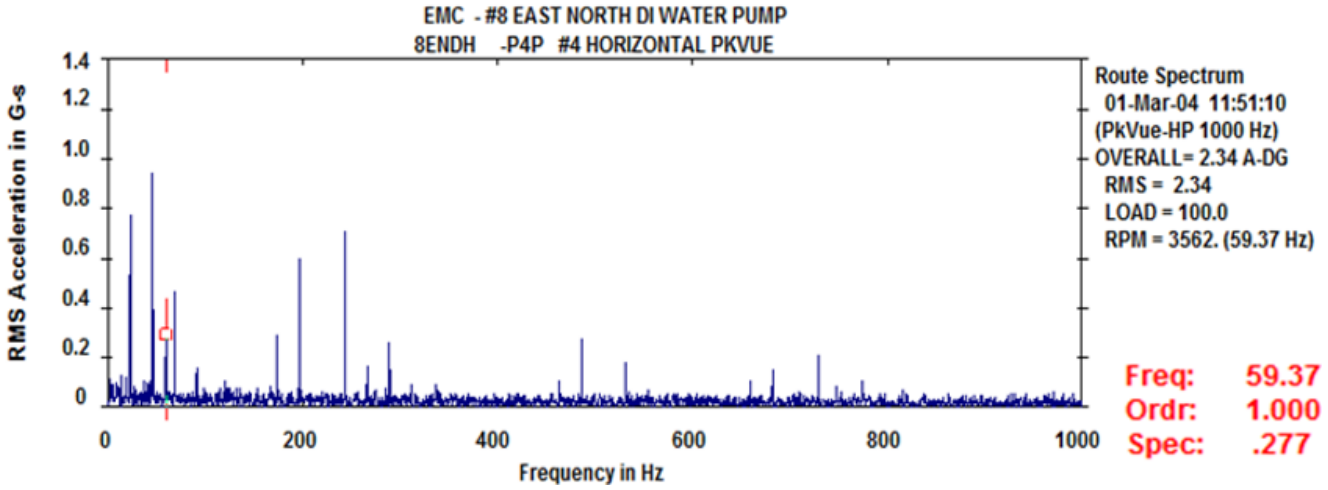
The maximum peak waveform amplitude in this example is 20.74 g's. This machine's alert and fault levels are shown. This amplitude is about twice the fault level for this machine, so the bearings should be scheduled for replacement at the next scheduled outage.



BSF

In this example, there is a peak in the spectrum at 1 x RPM. It should be noted that this peak is not due to imbalance.



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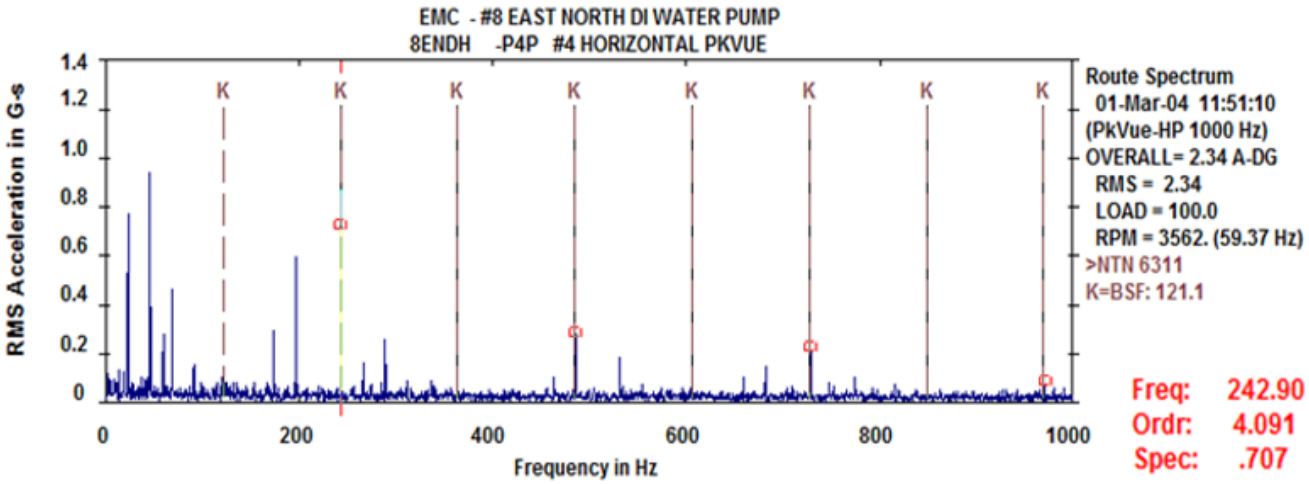
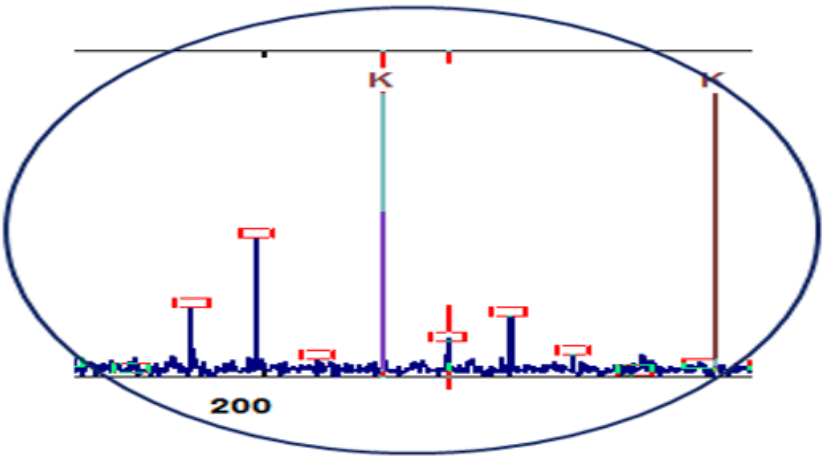
The cursor has been placed at the dominant peak above 1 x TS. Note that this frequency is equal to 2 x the calculated BSF. When a roller defect is present, it is common to see more amplitude at 2 x the calculated BSF because the defect impacts both race surfaces as it rotates within the bearing races.



An analysis of the sidebands around the peak at 2 x BSF reveals that the sideband spacing is equal to this bearing's FTF. Similar to conventional velocity spectral data, sidebands in the PeakVue spectrum are caused by modulation in the time waveform. This is a common pattern for roller defects since the defective roller goes into the bearing's load zone once per revolution of the cage or train.



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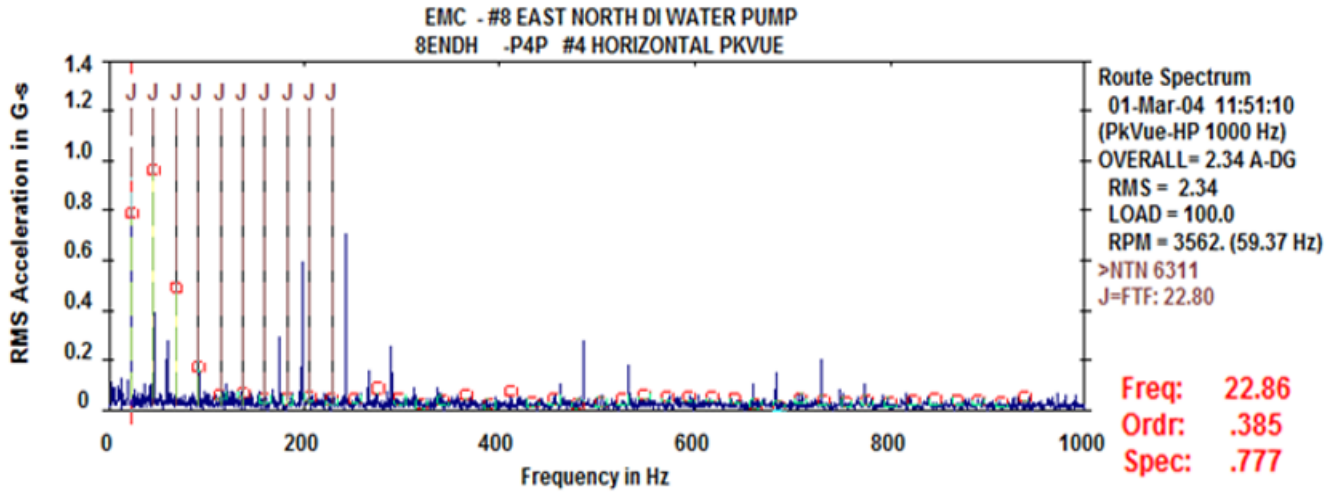
The two sub-synchronous peaks in the spectrum are also a result of modulation. The cursor identifies the first peak's frequency as .385 orders, and the second peak is a harmonic of the first. The fault frequency indicator identifies both of these peaks as FTF. This illustrates a big difference between PeakVue and traditional spectral analysis. In the traditional velocity spectrum, modulation frequencies are identified only by sidebands. In PeakVue analysis, modulation frequencies also show up in the form of sidebands, but modulation frequencies are also identified by the presence of discreet peaks and harmonics of the modulation frequency itself.



Lubrication Problems

Lubrication problems such as under lubrication, over lubrication, and even improper lubricant often result in the appearance of significant areas of broad band energy in the spectrum. Peaks may also appear in the spectrum with sideband spacing related to the bearing's FTF. Often, the first sign of a lubrication problem results in a significant increase in high frequency vibration amplitude. This is usually identified by an increase of either the HFD (High Frequency Detection of 5 kHz up to 20kHz) or the vHFD of 1kHz to 20 kHz parameters.

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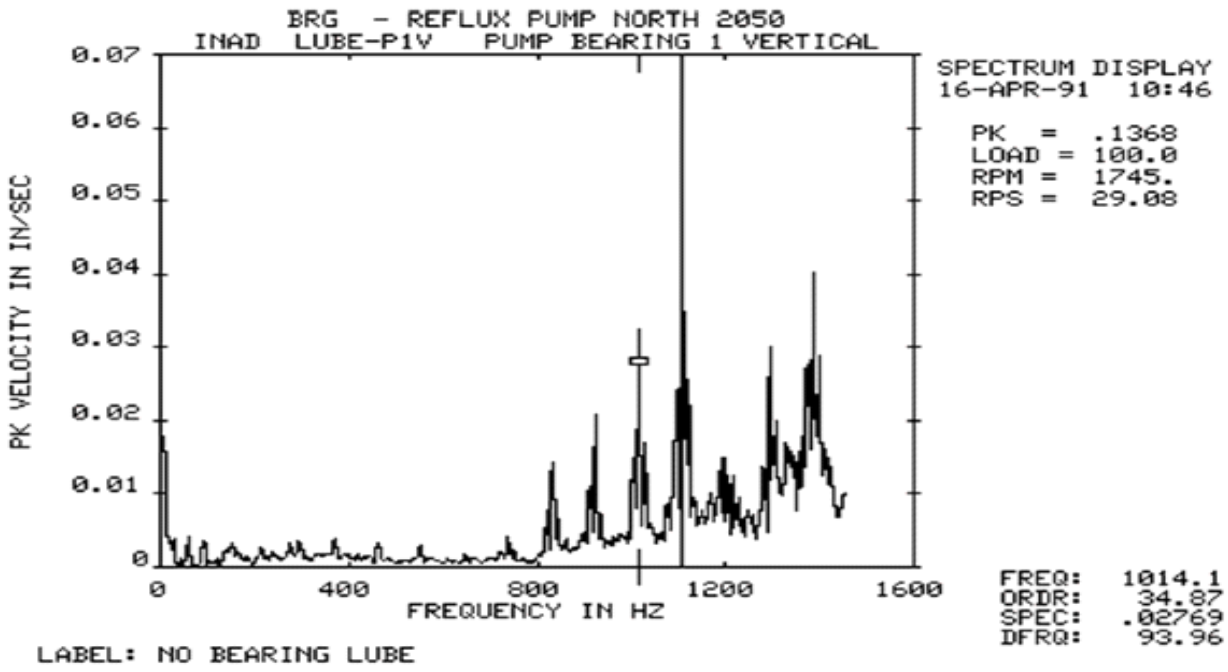
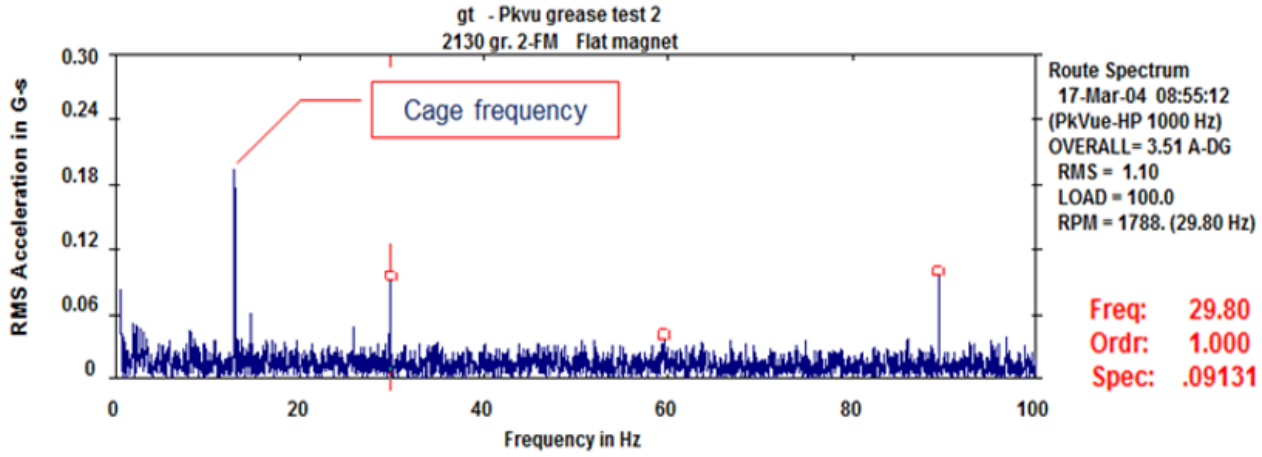


With PeakVue data, lubrication problems often result in the appearance of cage frequency (FTF) peaks in the spectrum. In this example, the cage frequency peak is actually the dominant peak. Other times, the PeakVue spectrum may have almost no frequency content at all.



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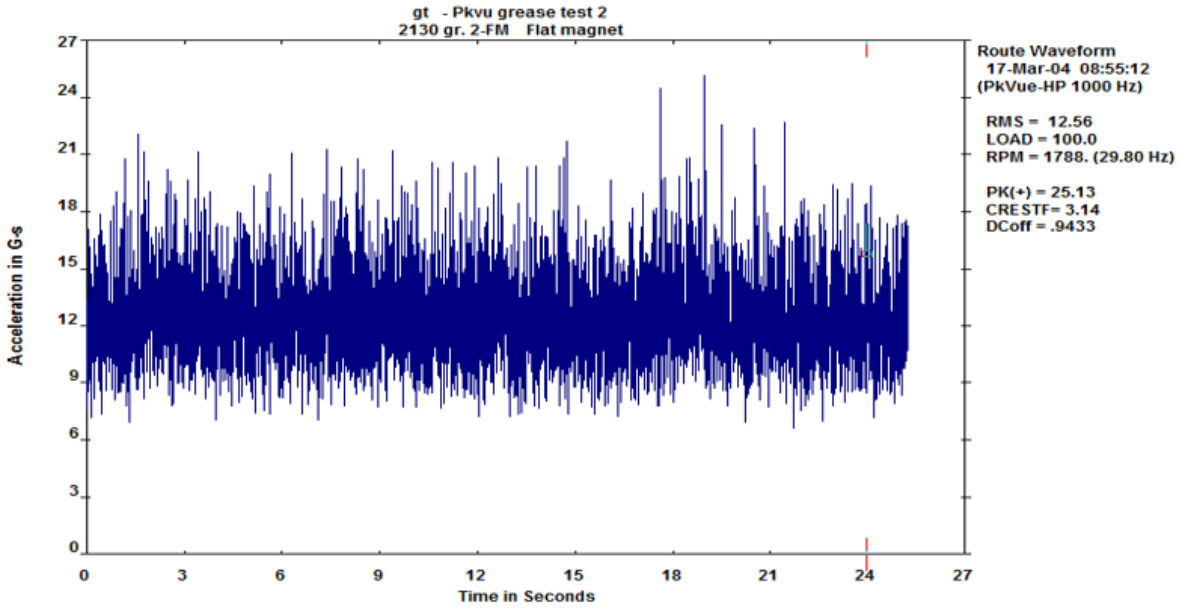
The PeakVue waveform will often reflect a very high amplitude with no obvious periodic signal. In this example, there is also a very high floor amplitude.



Lubrication problems should always be taken very seriously. Anti-friction bearings that are not properly lubricated may experience very rapid failure progressions. Also, prompt rectification of a lubrication problem may prevent further damage that could directly lead to a catastrophic failure.

Points to Consider

With traditional spectrum analysis, the appearance of a non-synchronous harmonic family almost always indicates the presence of an anti-friction bearing defect. Trend, spectrum and waveform data need to be analyzed when trying to determine the severity and type of defect before a proper recommendation can be made.

In most cases, PeakVue technology can detect anti-friction bearing defects earlier and make the analysis of those defects easier than with traditional vibration analysis. When analyzing PeakVue data, the spectrum is used to identify the type of defect, the waveform is used to gauge the severity.

In most cases, the best time to replace the bearing is when it has become unpredictable. There really is no dependable way to predict remaining bearing life after it has become unpredictable. As the bearing gets closer to failure, the risks related to unscheduled downtime and catastrophic failure increase exponentially.

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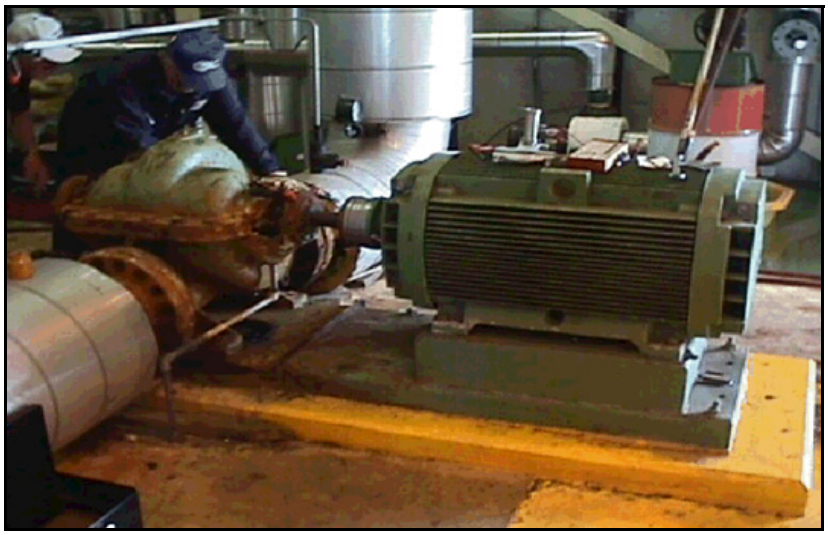
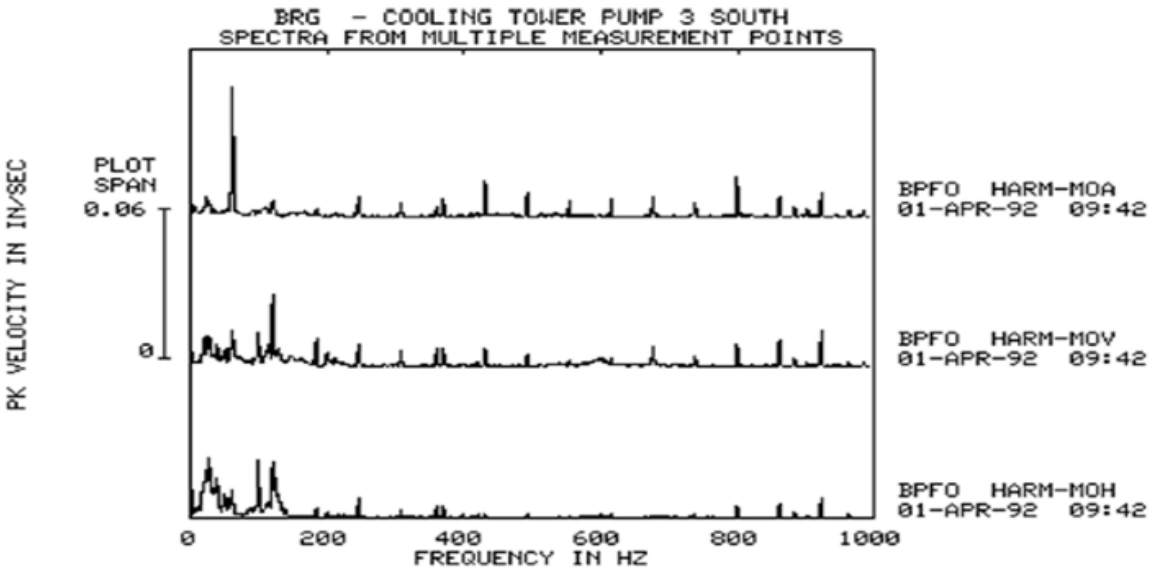
Case Histories



- 250 HP, 1200 RPM motor
- Motor has ball bearings.
- Most of the energy measured in the axial direction.

Spectra acquired in three directions of the motor outboard bearing are shown here. Notice that in this case, the vibration amplitude is highest in the axial direction. Also, notice that the spectral amplitudes are all relatively low.



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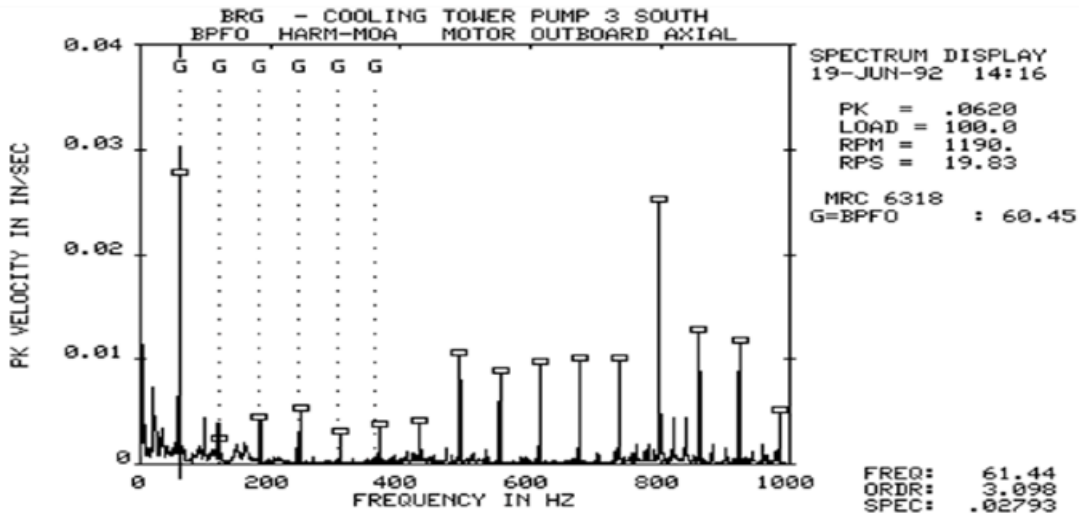
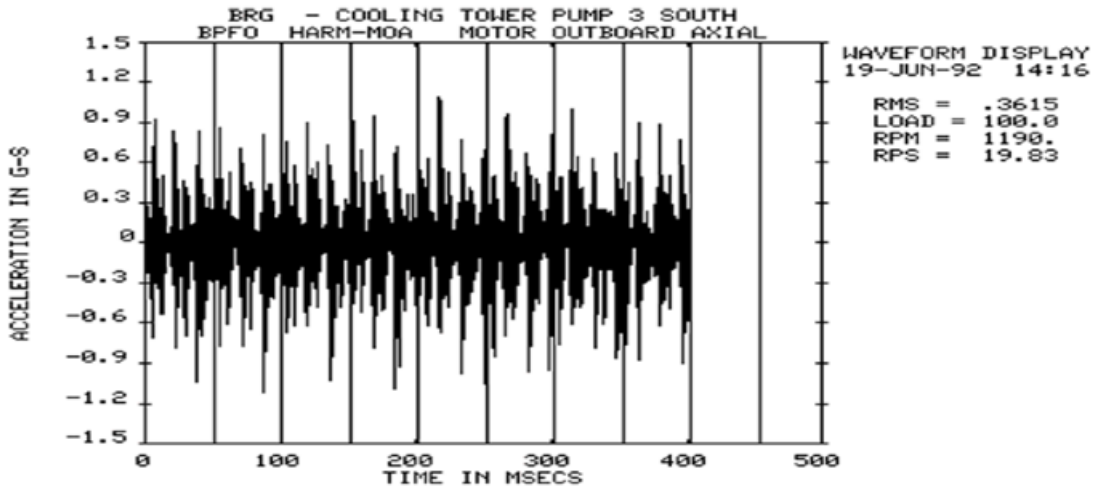
Looking at a single spectrum display of the motor outboard axial measurement point, shows that nearly all of the energy is harmonically related to the bearing BPFO. Based on this spectral pattern, the defect is in mode three of failure. The defining characteristic that leads to this conclusion is the obvious peak at 3.1 orders, which is this bearing's BPFO.



Viewing the waveform using acceleration amplitude units, gives the analyst a good indication of how much true impacting is occurring. In this example, the waveform amplitude is about 2 g's Peak to Peak. A trend of the waveform Pk-Pk value would be a very helpful piece of information in determining the proper course of action for this machine. Based on the given data, this bearing should be scheduled for replacement at the next scheduled equipment outage.



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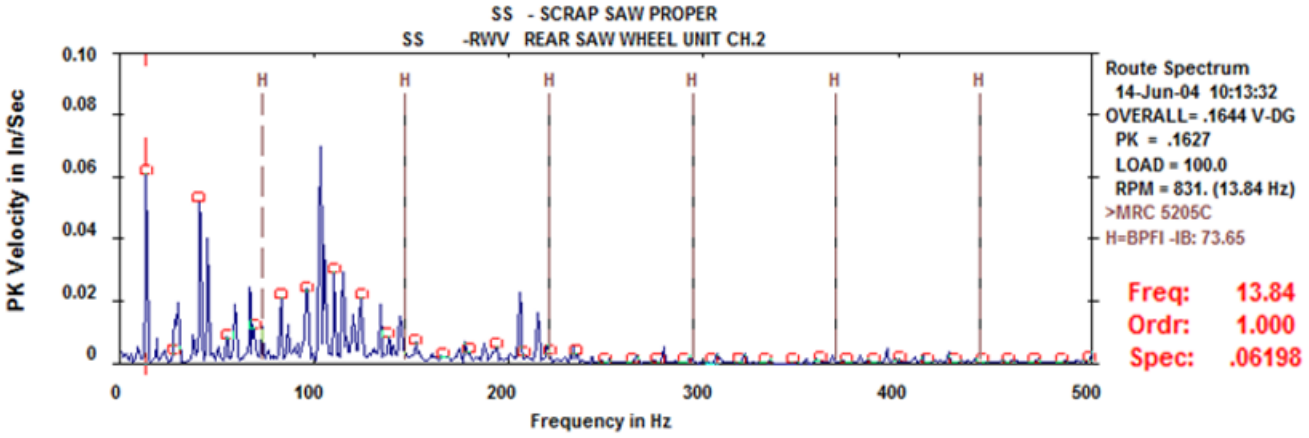
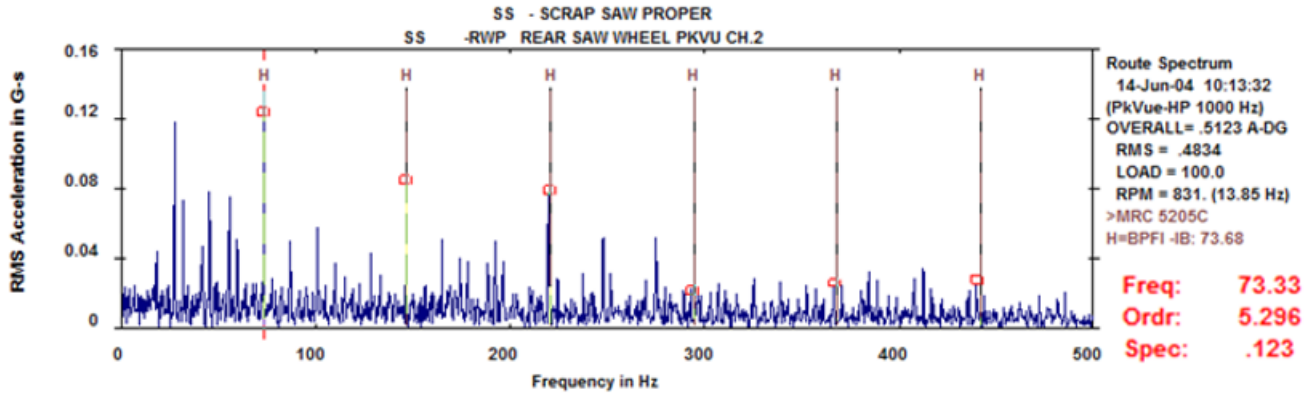
Scrap Saw Spindle Bearings

- 831 RPM
- Band Saw – data acquired while sawing 20” aluminum ingot

Conventional velocity vibration data does not appear to contain any bearing defect energy.



The PeakVue data acquired at the same measurement point at the same time shows that the dominant peak in the spectrum is at 5.3 orders, the BPFI of this bearing.



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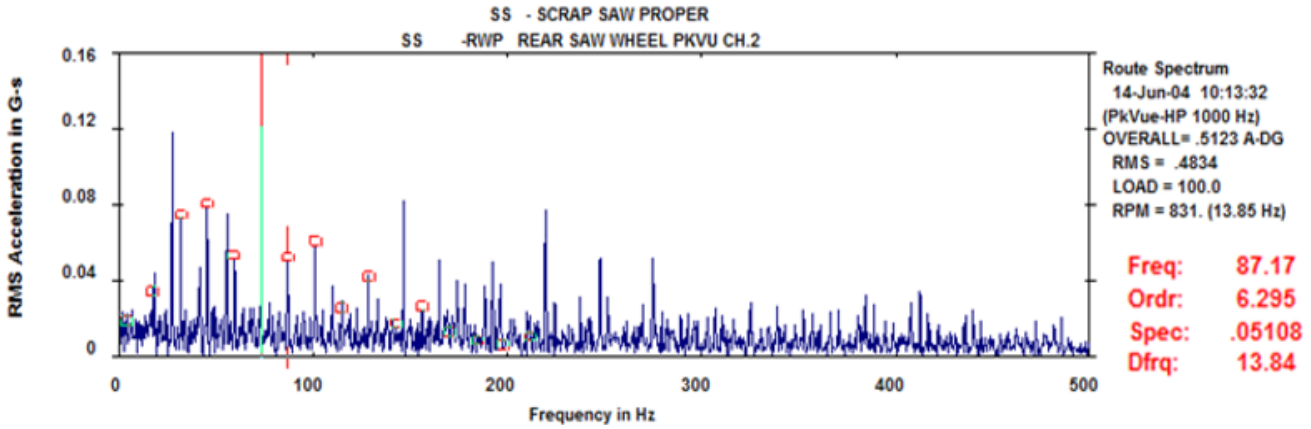
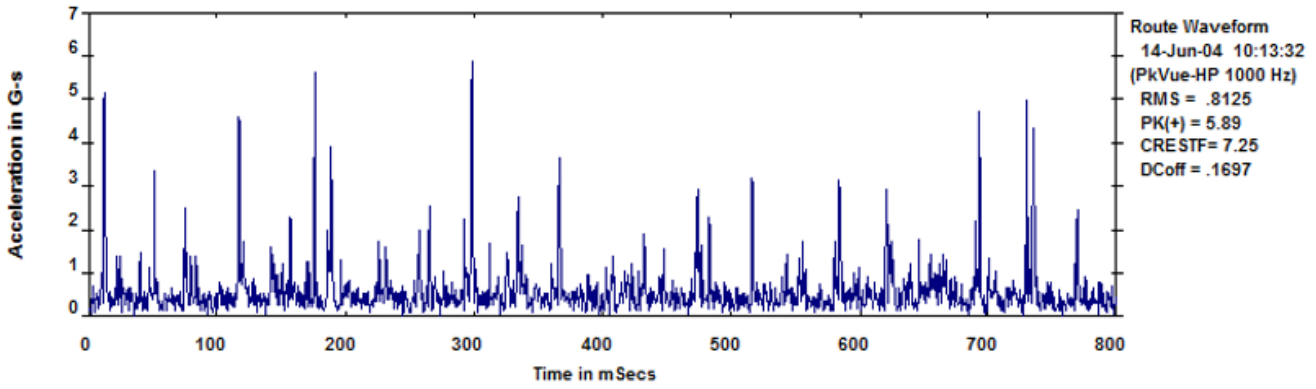
Sidebands appear around the peak at the calculated BPFI, which are spaced at 1 x RPM. This pattern, along with the peak in the spectrum at 1 x RPM, is caused by once per revolution amplitude modulation as the defect goes in and out of the load zone of the bearing.



The waveform amplitude is 5.9 g's, which for this type of fault would be this bearing's fault level.



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The bearing was replaced two weeks later. The spalled area of the race extends nearly the entire circumference of the bearing.

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Review Questions

1. Anti-friction bearing defects are almost always which type of vibration?

   A. Synchronous

   B. Non-synchronous

   C. Sub-synchronous

2. True or False? In the first stage of failure, a noticeable peak will be present at the calculated defect frequency in the velocity spectrum.

3. True or False? In the first stage of failure, a noticeable peak will be present at the calculated defect frequency in the PeakVue spectrum.

4. If a bearing's BPFO is added to its BPFI, the result will be equal to what?

5. True or False? Anti-friction bearing defects are identified by the presence of turning speed harmonics in the velocity spectrum.

6. True or False? The presence of cage frequency peaks in the spectrum usually indicates a defective or broken cage.

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7. It is common to see sidebands around the calculated inner race defect as well as the harmonics. What will be the difference frequency of the sidebands? Why?

8. The life of a bearing is most affected by the _____.

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Journal Bearings

Section 9

Objectives

- Describe the differences between journal and antifriction bearings.
- Discuss how to properly monitor journal bearings.
- Introduce the orbit plot.
- Define oil whirl and oil whip.
- Discuss different journal bearing designs.

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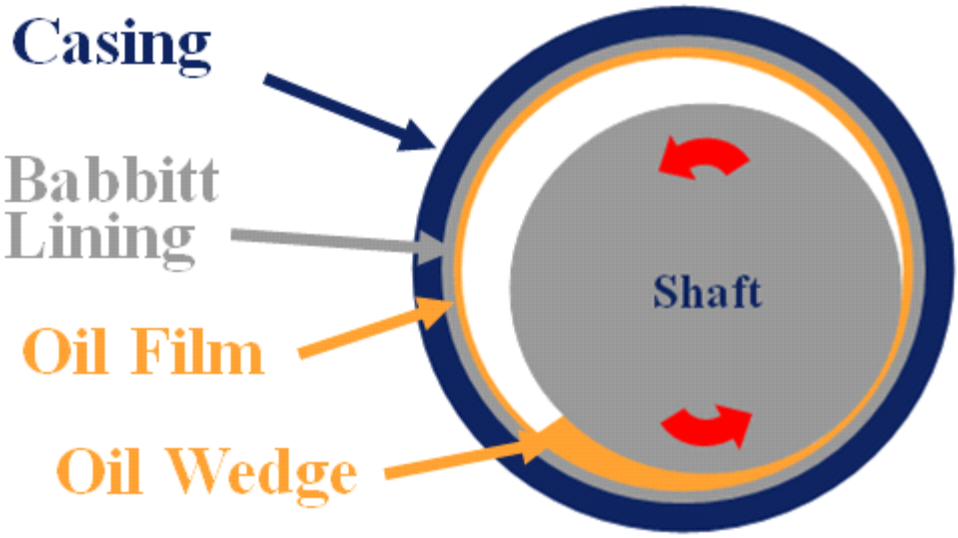
Introduction

An overwhelming majority of predictive technology is focused on machinery with antifriction bearings. However, many machines in industry are equipped with plain, or journal bearings. There are important differences that must be understood when performing vibration analysis on journal bearings.

These differences involve the type of sensor used for data collection, the method used for data collection and the way that data is analyzed. For this reason, the vibration analyst must always recognize whether a particular machine uses antifriction or journal bearings. This information is critical for performing vibration analysis on any machine.



Journal bearings consist of a shaft, or journal, which rotates freely within a supporting shell that is lined with babbitt. The term **babbitt** refers to the layers of softer metals (lead, tin and copper), which form the metal contact surface of the bearing shell. These softer metals overlay a stronger steel support shell and are used to protect the harder rotating shaft in the case of any inadvertent journal-to-casing contact.

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During normal operation, the babbitt does not support the shaft. Rather, it is supported by a film of oil, which supports the weight and the load of the shaft. When at speed, the shaft rotates fast enough to force oil between the curved surfaces of the shaft and the shell, creating an oil wedge and a hydrodynamic oil film. This film allows bearings to support even very heavy loads and very high rotational speeds. Since there is a designed internal clearance within the bearing between the shaft and the housing, movement that occurs within this clearance will not be measured by a casing mounted sensor, such as an accelerometer or a velocity transducer. For this reason, use of a displacement probe is preferred when acquiring vibration data from a journal bearing.

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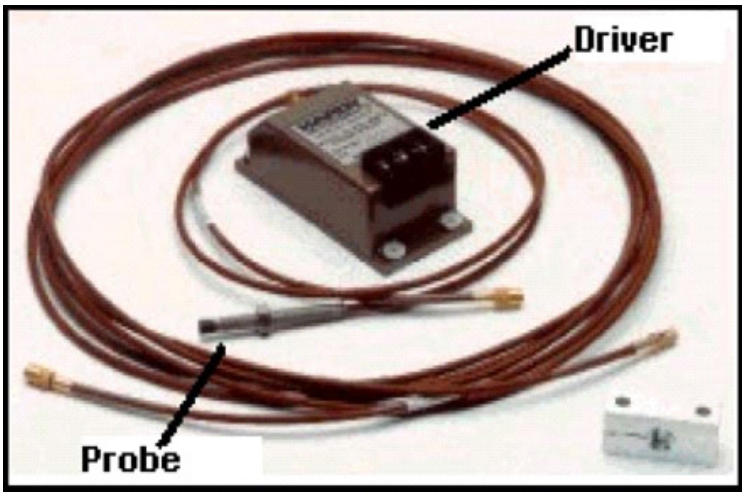
Displacement Probe

The **displacement probe**, which is sometimes referred to as an eddy current probe, uses an external electrical supply which, when directed to the probe tip generates a magnetic field. As it vibrates, the shaft passes through the magnetic field, which causes an electrical signal proportional to the vibration of the shaft.

Typical sensitivity of a displacement probe is 200 mV per mil with a gap voltage within the middle of the power supply source.



Another reason for the use of displacement probes is increased damping. In a journal bearing, there is much more damping than in a rolling element bearing due to the lubricant present. More viscous and thicker lubricant films provide higher damping properties. As the damping increases, the bearing stability also increases. A stable bearing design holds the rotor at a fixed attitude angle during transient periods, such as machine startups/ shutdowns or load changes.

The damping properties of the lubricant also provide an excellent medium for limiting vibration transmission. Thus, a vibration measurement taken at the bearing outer shell will not represent the actual vibration experienced by the rotor within its bearing clearances. However, it may be helpful to collect housing data to determine if there are any high frequency signals present. This could indicate impacting from the coupling, gears, the adjacent machine, etc.

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Advantages

- Measures the relative motion between the probe tip and the rotating shaft; ideal for machinery with journal bearings.
- Extremely useful when little vibration transmits to the machinery case.

Disadvantages

- Requires permanent installation, which often proves difficult and sometimes impossible.
- The frequency response is typically linear from DC to 1,000 Hz.
- Requires an electrical source, and signal conditioning affected by electrical runout.

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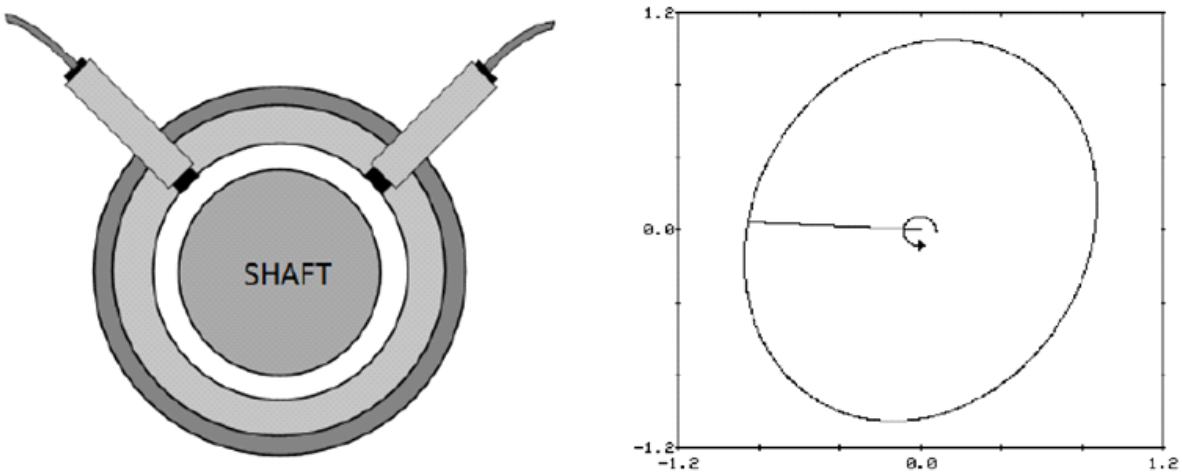
Vibration Analysis

The spectral Fmax for data taken from a journal bearing is relatively low, usually about 10 to 15 times shaft turning speed. High frequencies such as blade pass, gear mesh and cavitation would be attenuated by the lubricant, so they are not typically measured.

Case mounted sensors would not provide an accurate indication of these higher frequency vibrations, due to the inherent damping offered by the lubricant between the shaft and the bearing. The main vibrations that are analyzed are mainly due to the forces of imbalance, misalignment, resonance and looseness.

In the case of early wear in a journal bearing configuration, several harmonics of turning speed may be present. By design, journal bearings constitute a certain degree of looseness. A good baseline and trend data will help determine when the condition is excessive.



Usually, displacement probes are mounted at 45 degree angles from top dead center, 90 degrees apart at each bearing. When data is taken from both sensors at the same time, the resultant waveforms can then be combined into what is known as an **orbit plot**. The orbit plot represents the path of the shaft centerline as it rotates within the bearing. It can be analyzed to detect abnormal movement of the shaft, which may be caused by a developing fault.

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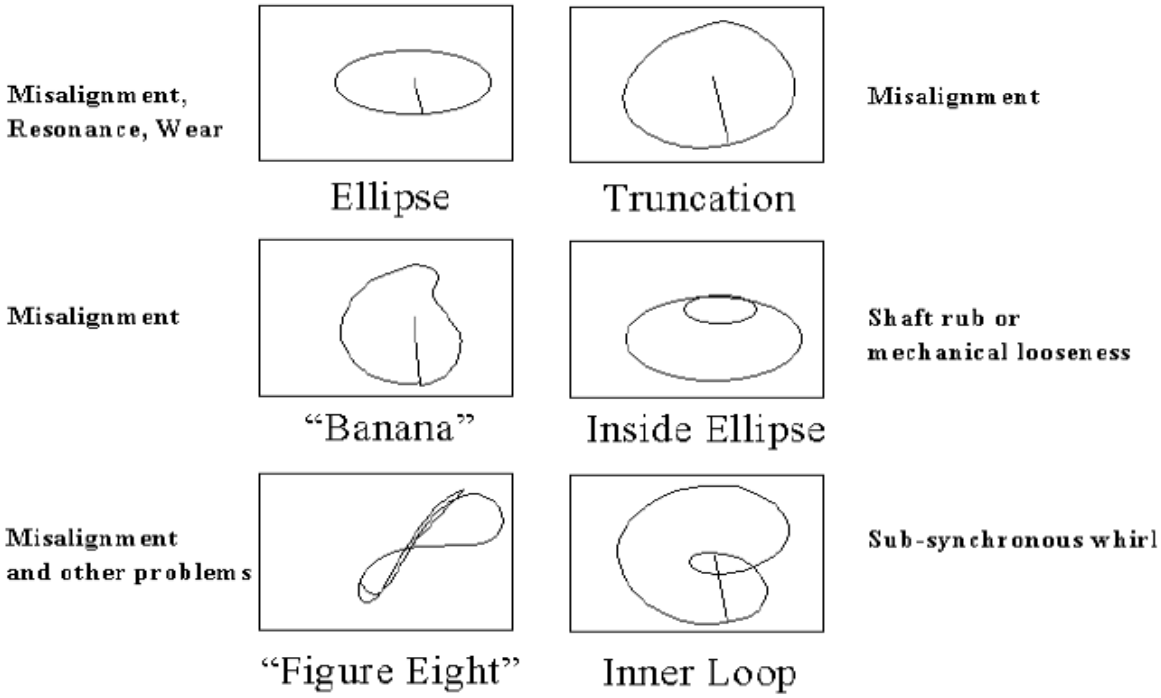
If performed correctly, orbit plots may provide additional insight into the condition of the machine. This illustration displays different characteristics of typical faults.



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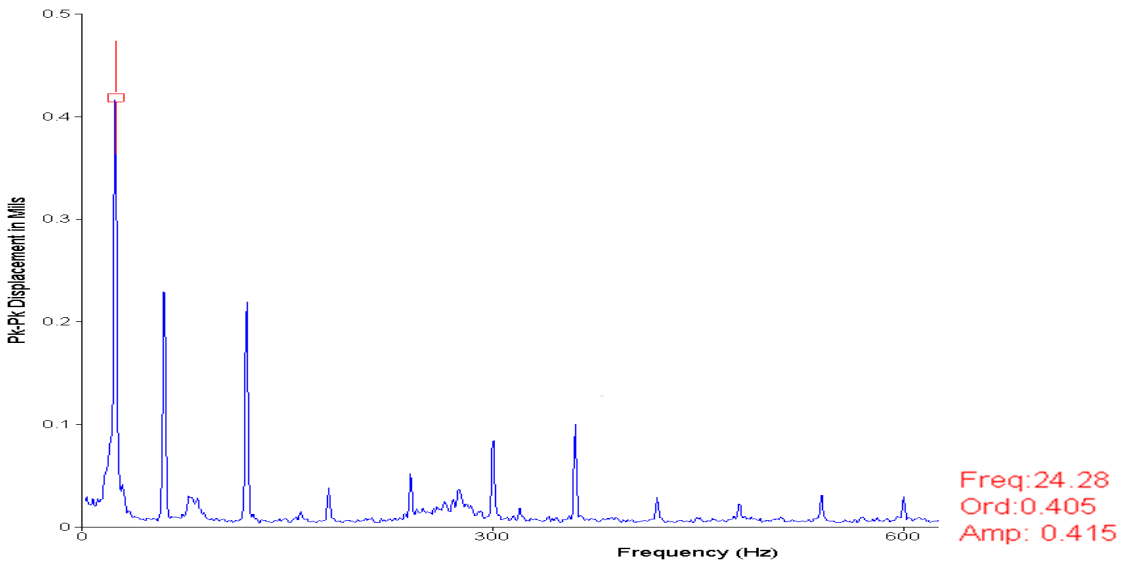
Oil Whirl / Oil Whip

Oil whirl is a phenomenon that can occur in journal bearings when the shaft position within the shell becomes unstable, and the shaft continues to change its position during normal operation due to the fluid forces created within the bearing. Oil whirl may be reduced by increasing the load or changing the viscosity, temperature or oil pressure in the bearing. A permanent solution may involve a new bearing with different clearances or design. **Oil whip** occurs when the oil whirl frequency coincides with the system's natural frequency. The result can be a catastrophic failure.



In the spectrum, oil whirl is seen as a sub-synchronous peak between .4 and .48 orders of shaft turning speed. The amplitude may be very erratic, so Peak Hold averaging is recommended to measure the true amplitude of the peak.

Protection

Oil whirl, oil whip or loss of lubrication can cause catastrophic machinery failure very quickly and unpredictably. Due to the unpredictable nature of journal bearings, companies commonly install continuous monitoring and protection systems on critical machinery that use journal bearings.

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Often, it is through these systems that the vibration analyst is able to gain access to the displacement probes for periodic vibration data collection. When this is the case, the analyst must make sure to first get permission from operations before attempting to connect anything to a protection system.

Also, the analyst needs to ensure that he or she knows exactly which types of probes and connections are being used, and that the analyzer is set up properly for connection to that system. The wrong analyzer set up connected to the wrong type of connection could possibly cause an inadvertent machinery trip, which could take hours, days or even weeks to rectify.

Before connecting the analyzer to a protection system make sure to:

- Get permission and notify control personnel that you are connecting the analyzer to the protection system.
- Identify the type of sensor that is being used. Do not assume that each sensor in the protection system is a displacement probe.
- Turn the analyzer's sensor power off, even when connecting to an accelerometer. Any sensor that is being monitored by the protection system is being powered either by an external driver or by the monitoring system directly.

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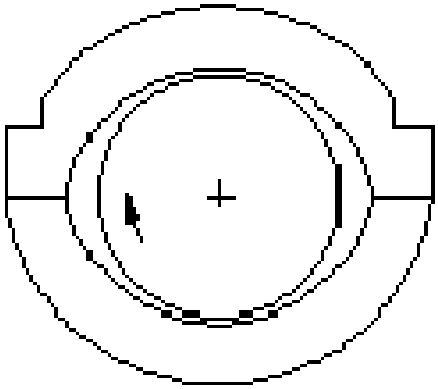
Journal Bearing Types

Since oil whip can be so catastrophic, bearing manufacturers have different types of journal bearings designed specifically to prevent oil whip. Some common designs employed are lemon bore, pressure dam and tilt pad bearings. These designs were developed to interrupt and redirect the oil flow path within the bearing to provide higher bearing stabilities.

Lemon Bore

The **lemon** or **elliptical bore** bearing is a variation on the plain bearing where the bearing clearance is reduced in one direction. Shims are installed during manufacture of this bearing at the split line, and then it is bored cylindrical. When the shims are removed, the lemon bore pattern is the result.

For horizontally split bearings, this design creates an increased vertical pre-load onto the shaft. This bearing has a lower load carrying capacity than plain bearings, but they are still susceptible to oil whirl at high speeds. Manufacturing and installation costs are considered low.



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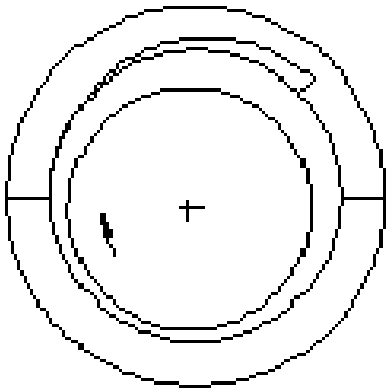
Pressure Dam

A **pressure dam** bearing is similar to a plain bearing which has been modified to incorporate a central relief groove along the top half of the bearing shell, which ends abruptly at a step.

As the oil is carried around the bearing, it encounters the step which causes an increased pressure at the top of the journal inducing a stabilizing force onto the journal. This forces the shaft into the bottom half of the bearing. This bearing has a high load capacity, and is a common correction for machine designs susceptible to oil whirl.

Pressure dam bearings are a unidirectional configuration.



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Tilting Pad

The **tilting pad** bearing is a partial arc design. This configuration uses individual bearing pads which are allowed to pivot, or tilt, to conform to the dynamic loads from the lubricant and shaft.

This type of bearing is a unidirectional design, and is available in several variations incorporating differing numbers of pads with the generated load applied on a pad or between the pads.

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Review Questions

1. Oil Whirl is which type of vibration?

   A. Synchronous

   B. Non-synchronous

   C. Sub-synchronous

2. True or False? The failure stages of a journal bearing are nearly identical to those of an anti-friction bearing.

3. True or False? The Fmax used when acquiring data on a journal bearing machine is typically the same as with an anti-friction bearing machine.

4. At what locations are displacement probes typically installed on journal bearings?

5. True or False? An orbit plot can be acquired with a single-channel analyzer.

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Looseness

Section 10

Objectives

- Define looseness.
- Describe two types of looseness.
- Identify spectral, waveform and phase characteristic(s).
- Determine corrective actions.

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Looseness

Looseness is the state or condition of a machine where its fasteners or rotating element are no longer held fast or rigid to its host. It is often classified as one of two types.

Structural Looseness:

- Base Mount
- Split Casings
- Bearing Caps
- Bearing Supports

Rotating Element Looseness:

- Impellers
- Fans
- Bearings
- Couplings

Fasteners, hold down bolts, etc. may work loose due to excessive or inherent vibration causing structural looseness. Also, grouting and other supporting mechanisms can be compromised over time causing a looseness condition.

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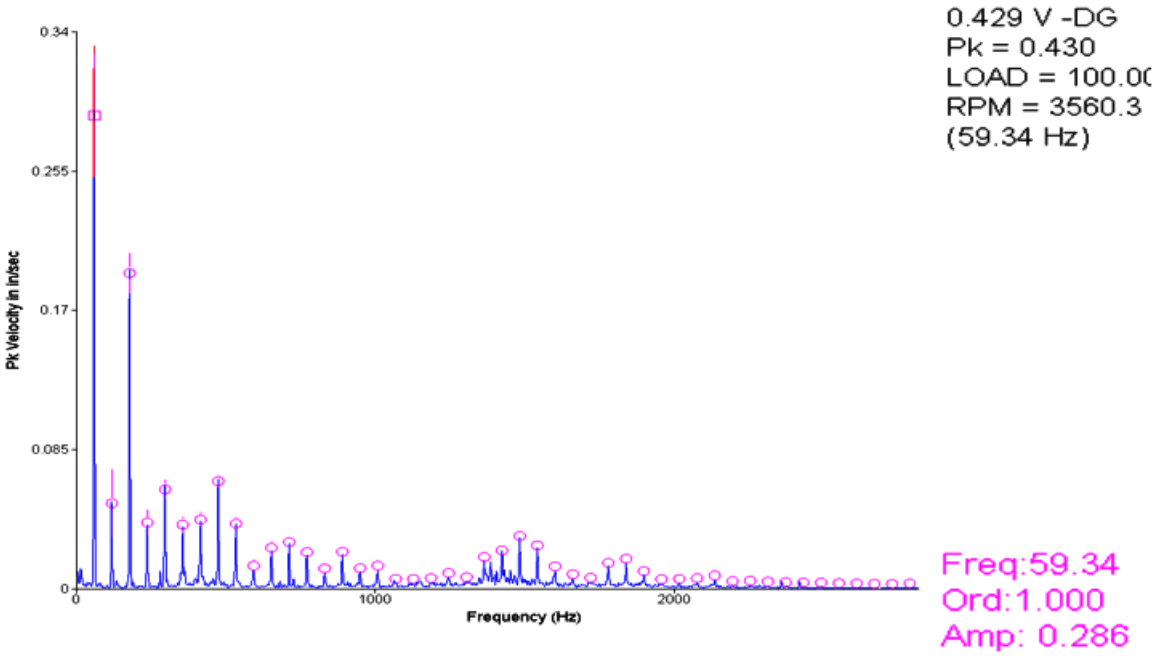
Characteristics of Looseness

Spectral

Looseness is identified in the spectrum by the appearance of multiple harmonics of the shaft turning speed (1 x TS), often accompanied by a raised noise floor at the base of the harmonics. The spectral pattern is the same regardless of which type of looseness is present, which makes identifying the type of looseness very difficult through vibration analysis alone.

If the problem continues, the spectrum may exhibit half order harmonics in between the 1 x TS harmonics such as 0.5, 1.5, 2.5 orders etc.

On some machines, multiple harmonics of turning speed are quite normal. Machines that have a linear motion such as diesel engines, reciprocating compressors, milling machines, etc. will often generate multiple turning speed harmonics due to the back and forth motion inherent in their normal operation.



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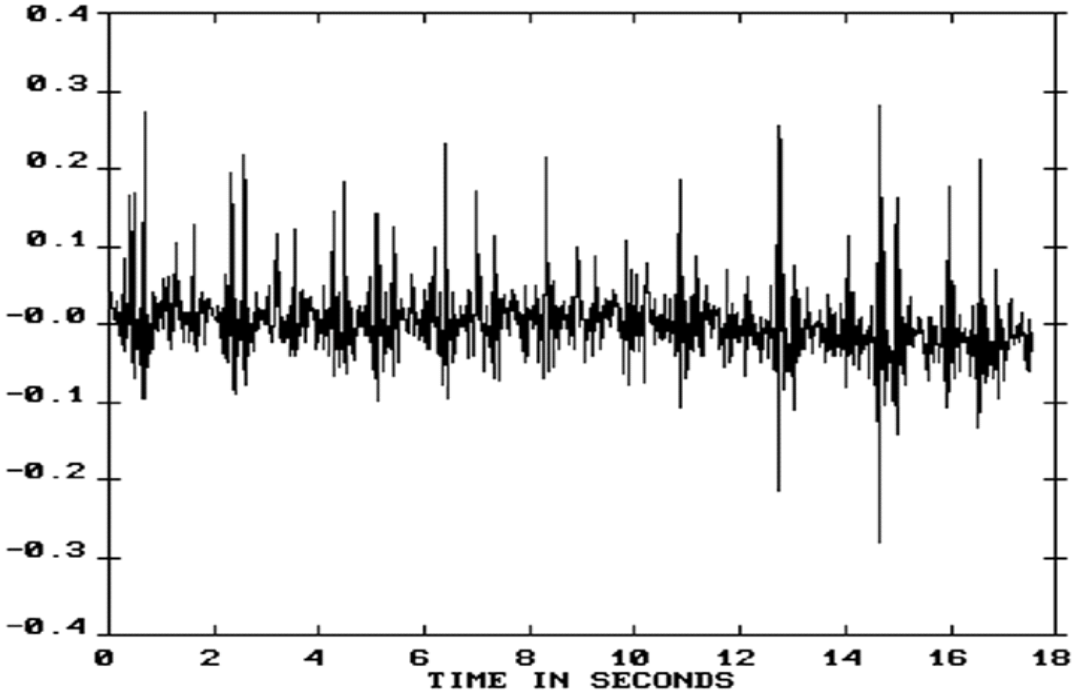
Waveform

The most common word used to describe the waveform pattern associated with excessive looseness is **random**. Looseness is known to produce waveform signatures that are often asymmetrical, truncated and random in appearance. Impacting will appear as a result of the loose component moving away from and then contacting the housing or base.



Phase

The word random is used often to describe looseness because of what happens when phase is measured on a loose component. In most cases, when phase is measured on a component where a significant amount of looseness is present, a phase value cannot be identified since it is not constant. The phase tends to be very random, which makes balancing a machine that suffers from excessive looseness very difficult, if not impossible. It should be noted that although the spectral pattern generated by looseness appears to be synchronous in the spectrum (multiple harmonics of 1 x TS), it is not truly synchronous because it is not phase locked with the RPM of the shaft.

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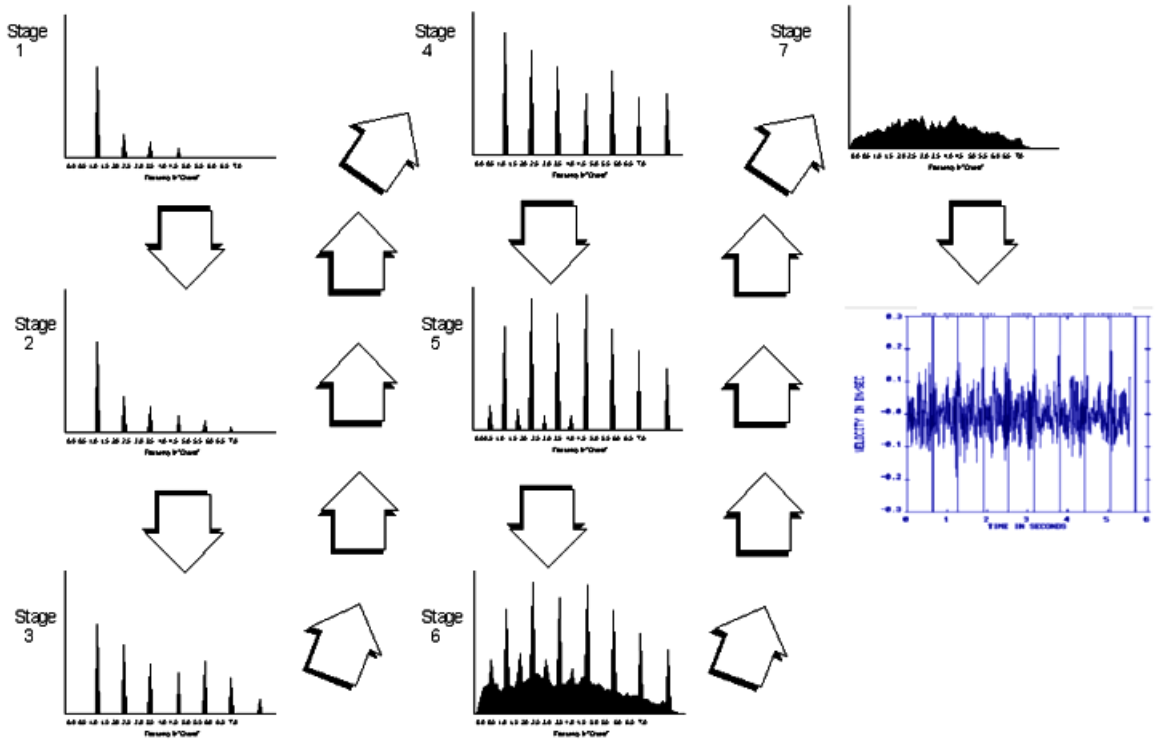
Identifying Severity

The determination of exactly how much excessive clearance exists in the component is very difficult, since the force of the impacting is dependent on not only the amount of clearance, but also on the amount of imbalance. A machine might have minimal clearance, but a large amount of imbalance force may result in severe impacting. On the other hand, a machine may have excessive clearance, but if the rotor is balanced there may be minimal impact force. For this reason, spectral pattern identification of different stages of looseness should be recognized.



As illustrated above, most machines will show a moderate amount of looseness as in stages one and two. As clearance increases, the pattern should result in more and higher amplitude turning speed harmonics shown in figures three and four. The appearance of half harmonics and a significant raised noise floor, as in stages five and six, indicates that the impacting has now progressed to a significant amount of rubbing. If not corrected, this could lead to stage seven, catastrophic failure.

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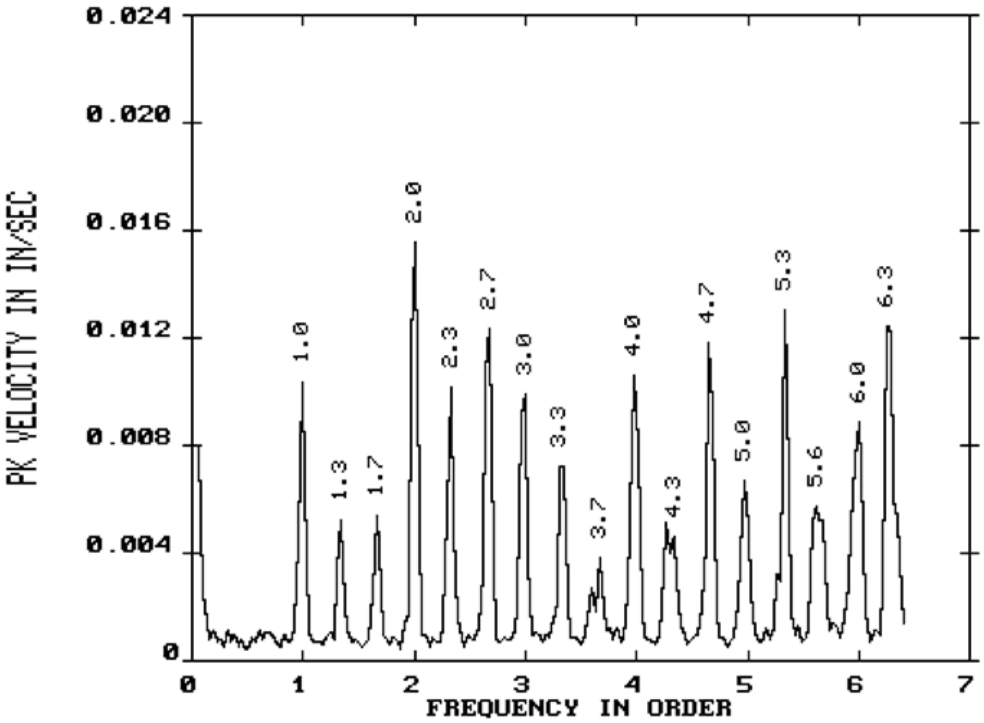

Fractional turning speed harmonics (in this example at 1/3 x TS) are common when a significant amount of rubbing is present.

Since it is so difficult to identify the true amount of excessive clearance in the system, physical inspection is the only reliable way to identify the severity of the problem. Once vibration analysis has identified the condition, the machine should be inspected. Begin by testing the integrity of the hold down bolts. If they can be loosened by hand, they need to be properly secured. Visually inspect the machine feet, fasteners, base plate and base for signs of looseness. If no structural problems can be found the only remaining option is to take the machine out of service and open and inspect the bearings and rotating assembly.

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Case History #1 - Pump Driven by Motor

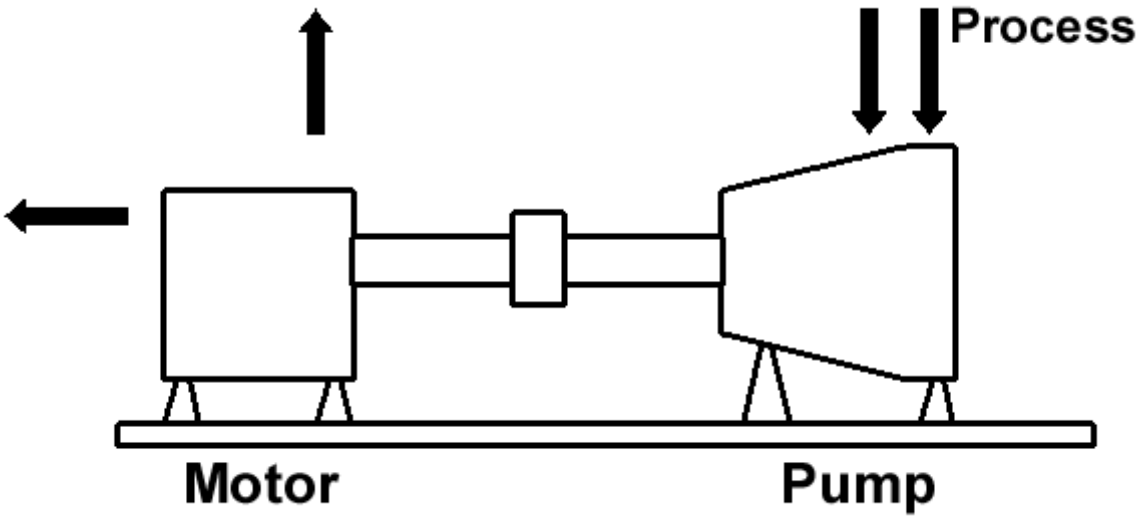

Figure 1

This case history illustrates a looseness fault as it develops over an eight month period. A brief history is given. There is no waveform to analyze.

April	Significant turning speed peak with no appreciative harmonic activity.
May	Broad-band energy around elevated 1x and 3x TS peak.
June / July	Many harmonics of turning speed now appear.
August	Over amplitudes increasing.
September	No data collected.
October	Fractional TS harmonics appear.
November	Fractional harmonics still present, scale increases to 0.6 IPS
December	Sharp increase in peak amplitudes; overall exceeds 1.0 IPS

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Elevated 1x TS, but no significant harmonics, indicating a probable imbalance problem. Some looseness is evident by the presence of low amplitude TS harmonics.

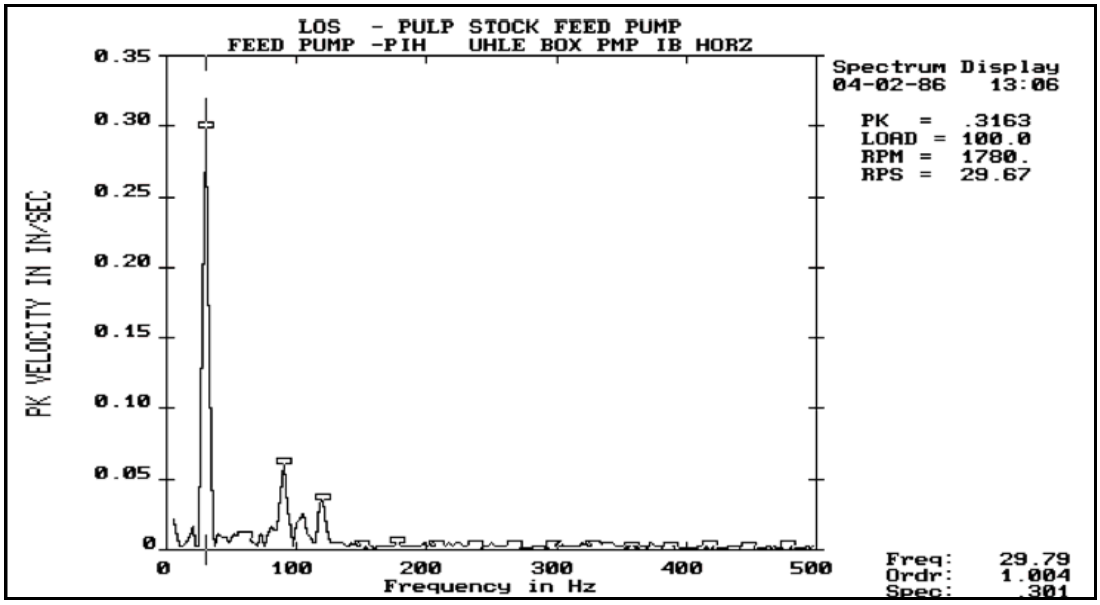

Figure 2

The data from May reveals that the 1 x has actually reduced in amplitude. The overalls have also decreased. The second significant peak appears to be 2 x TS. Closer inspection reveals that it is actually 3 x. The broad skirt around 3 x is a good indicator of looseness. (See Figure 3)

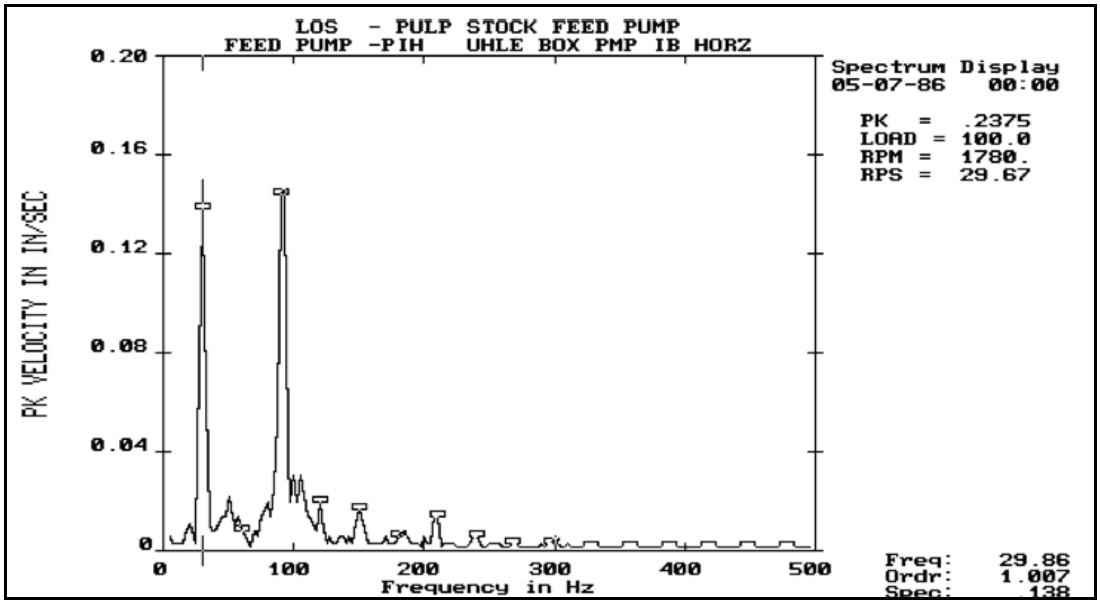

Figure 3

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By June, the overall vibration has diminished. The reduction of 1 x and harmonic amplitudes makes the machine appear to be healing itself. If amplitude were the only parameter being considered, a serious fault could go undetected.

Notice how the energy is being spread out among the harmonics of turning speed. Broad based energy is a strong indicator of equipment damage. (See Figure 4)

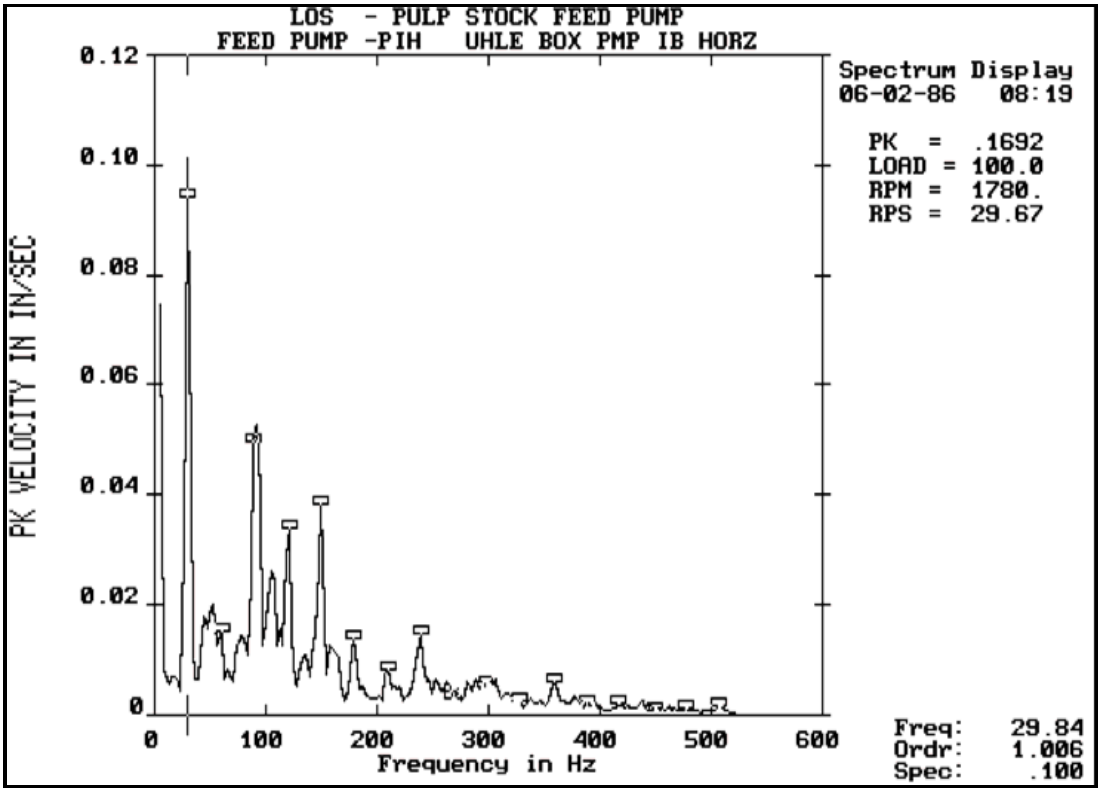

Figure 4

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Turning speed amplitude has not changed much by July. Overall energy has nearly doubled. More harmonics have appeared. The bases of all peaks have broadened. This broadband energy indicates a serious problem. (See Figure 5)

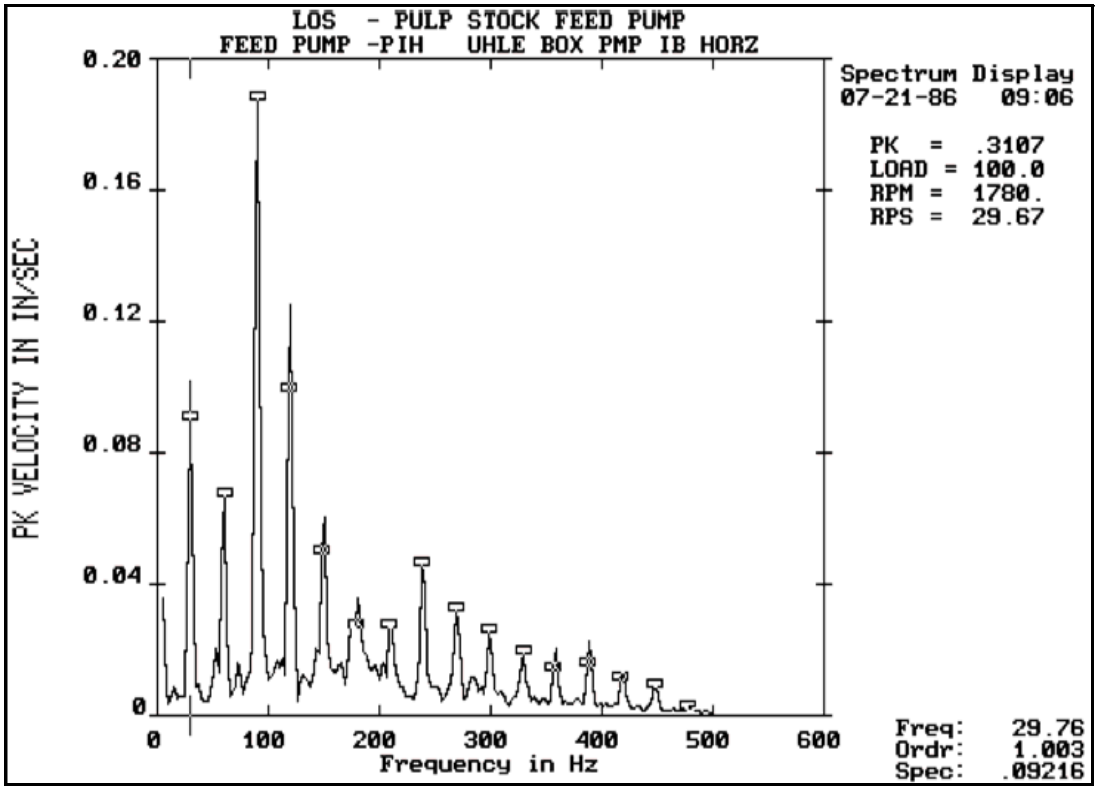

Figure 5

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The scale has more than doubled. Overall energy has nearly doubled from last month. Initially, the broadband energy appears to have disappeared. However, the scale is so high that those amplitudes are somewhat suppressed, giving the appearance that the problem is not as serious as previously thought. (See Figure 6)

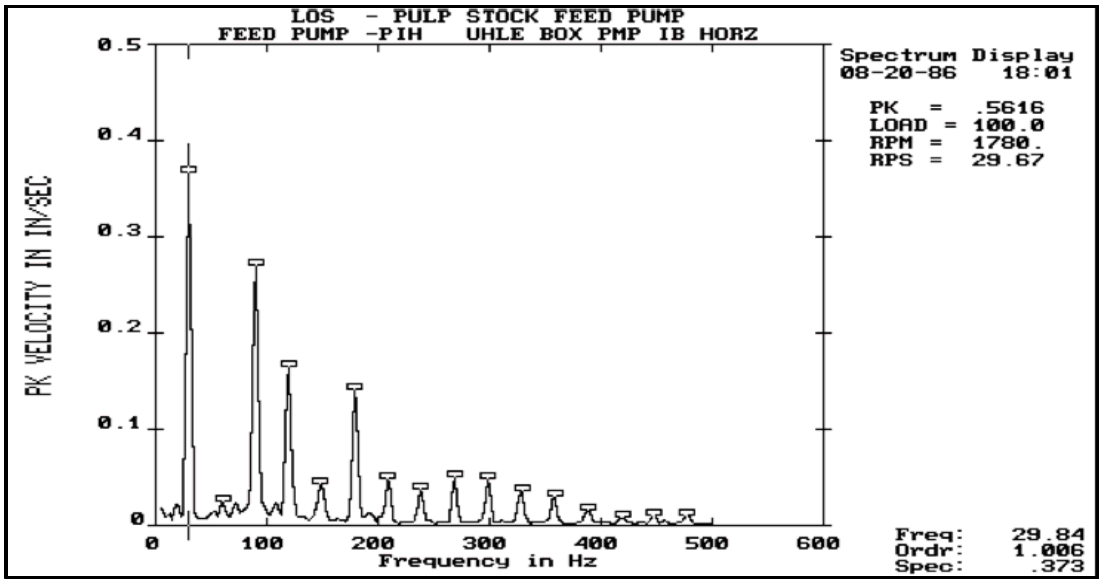

Figure 6

Two months later the scale is reduced by a factor of more than three. Overalls are nearly half of what they were. Harmonics up to 15 x are present with a raised noise floor. (See Figure 7)

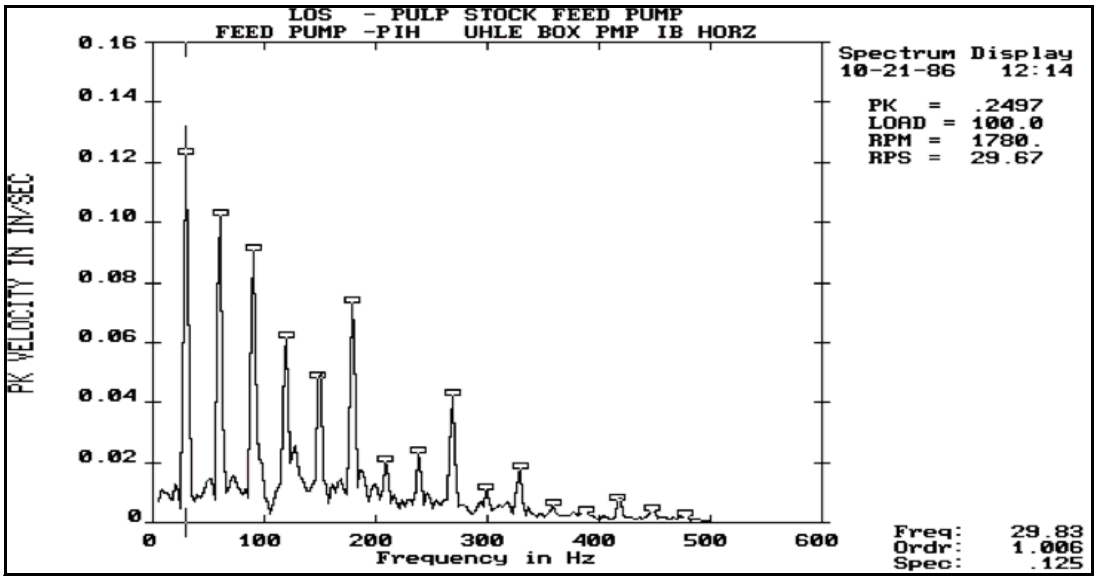

Figure 7

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1 x and overalls have increased by a factor of five by November and fractional harmonics are present. The presence of fractional harmonics indicate a problem, regardless of amplitude. (See Figure 8)

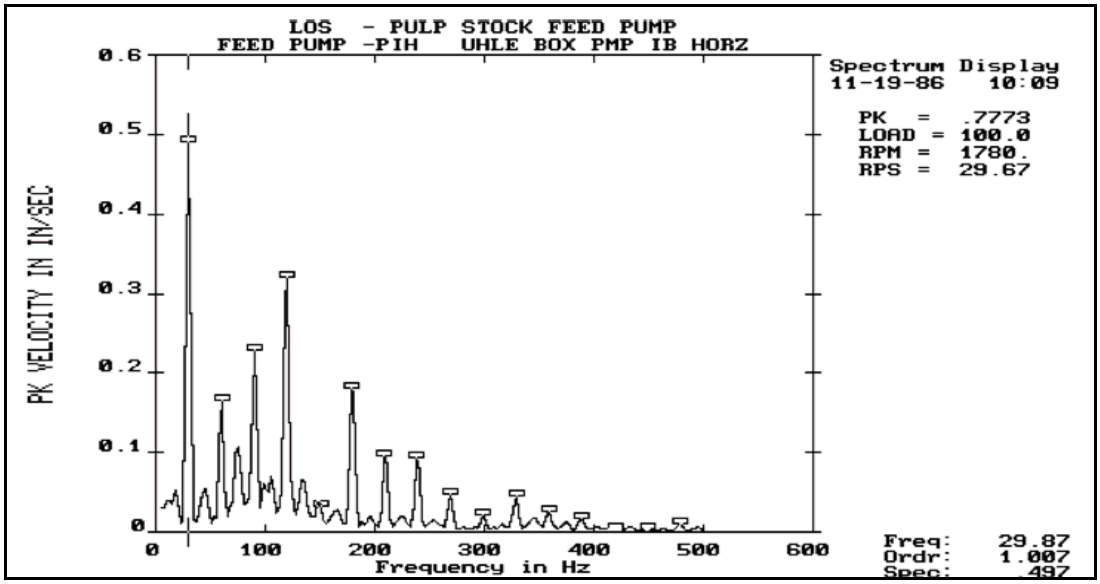

Figure 8

Turning speed is the dominant peak, with harmonics out to the Fmax. Overalls exceed 1in/sec. Catastrophic failure is imminent.

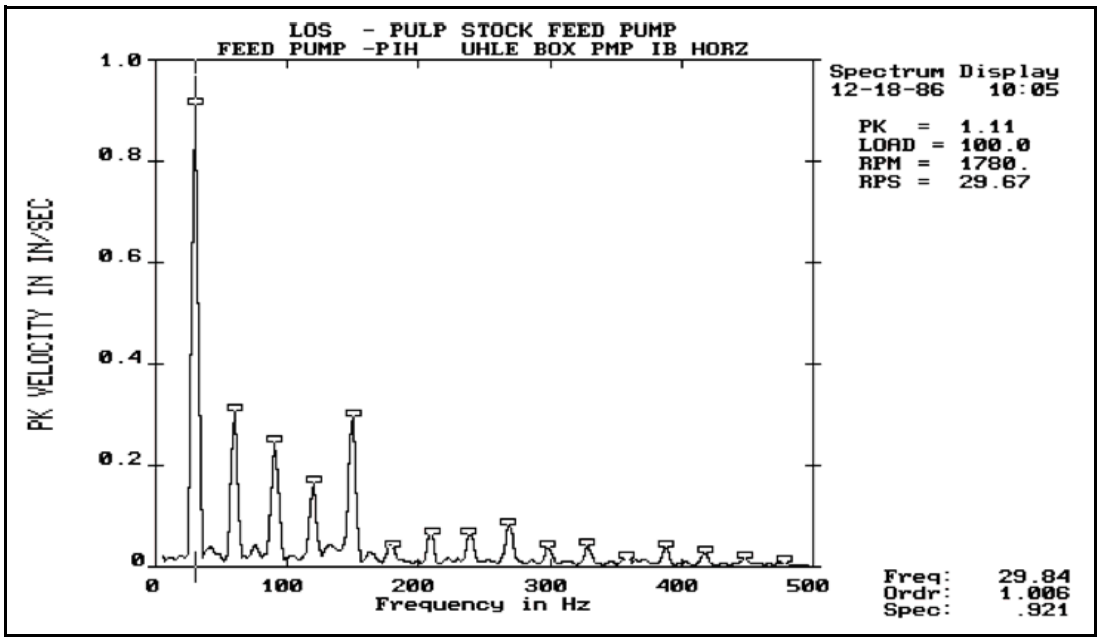

Figure 9

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The trend of overall energy shows an erratic pattern. This usually indicates severe changes in structure integrity, whether in the mounting fixtures or the container of the rotating element in question. What began as a simple imbalance problem has advanced to a major rebuild and costs are multiplied considering the man-hours, material and loss of production due to downtime. (See Figure 10)

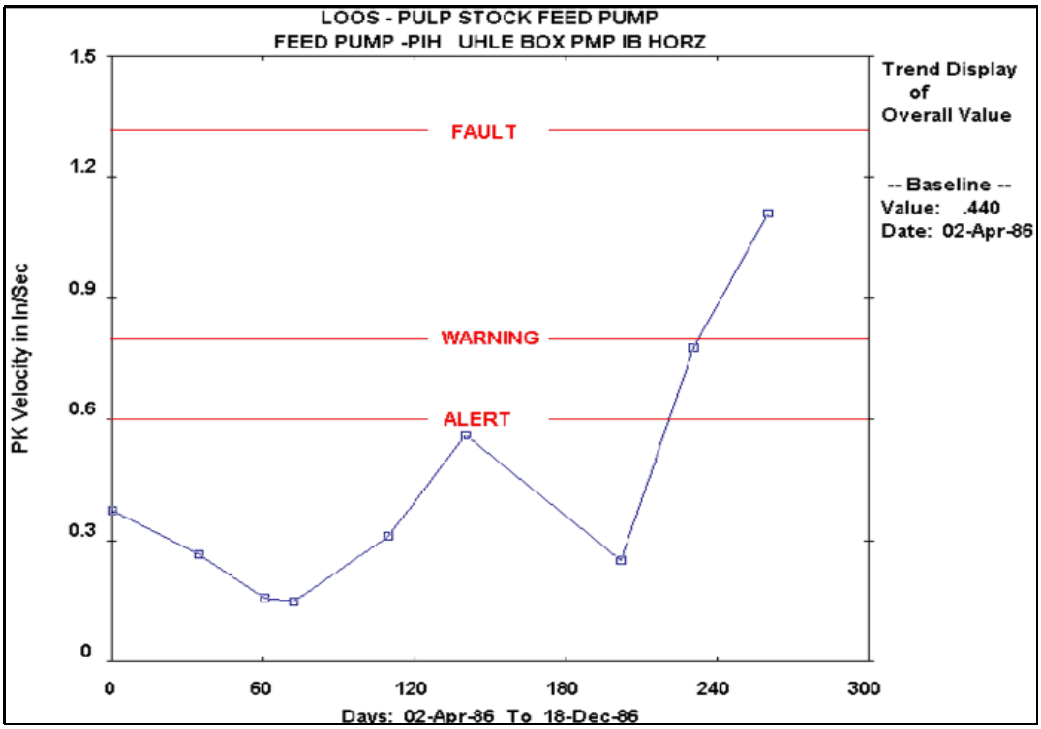

Figure 10

Data Analysis

- Dominant turning speed with harmonics
- Broad band energy with fluctuating overalls
- Strongest in the radial directions.

Diagnosis

Structural looseness, either in the base or the rotating element housing.

Corrective Action

Balance the equipment, inspect base for loose fasteners and inspect housings for damage.

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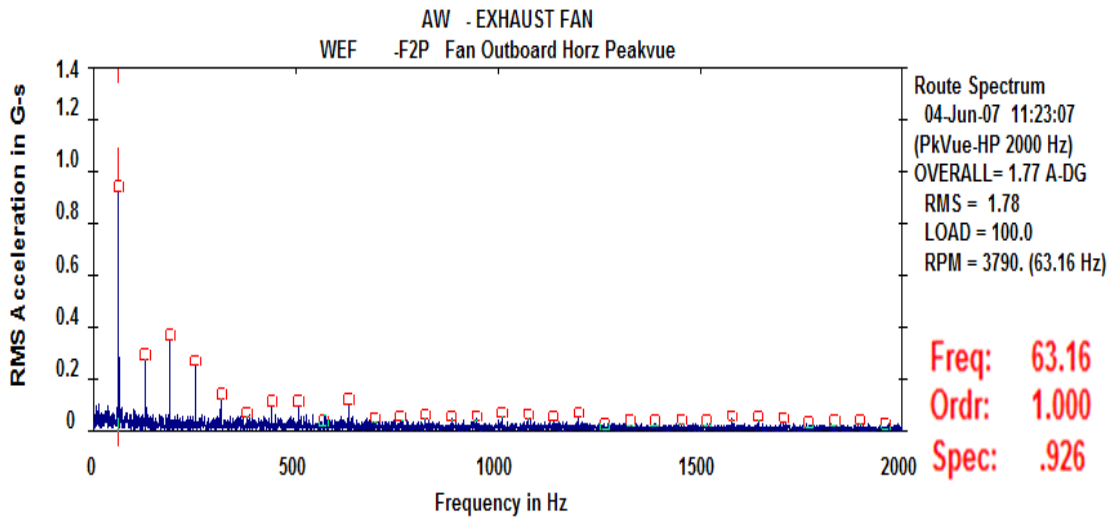
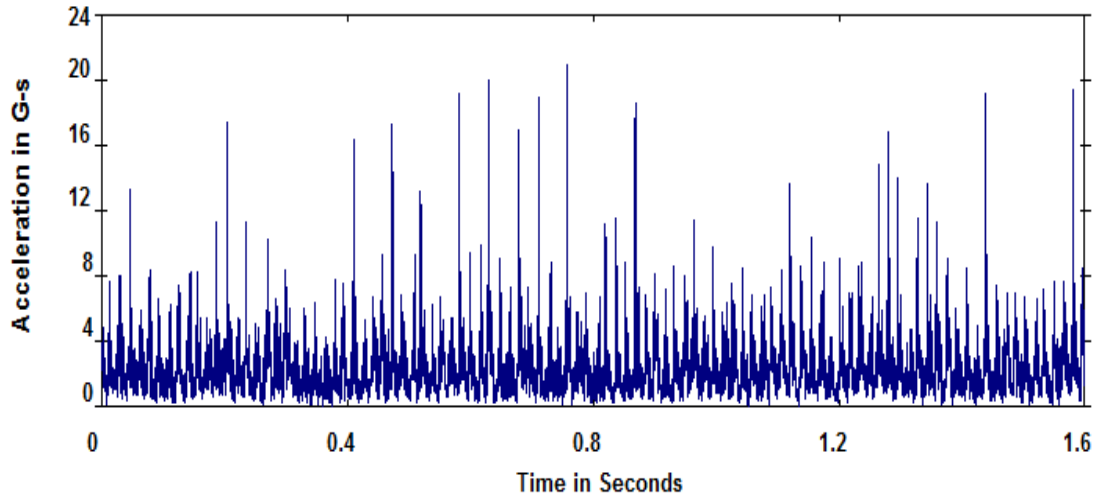
Case History #2 - AW Exhaust Fan

An example of looseness identified using PeakVue. Data was acquired using PeakVue, 2000 Hz Fmax, 2000 Hz High Pass filter and 3200 LOR.



In PeakVue data, the presence of peaks at 1 x TS and harmonics indicates that stress waves are being generated at 1 x TS. Note the random pattern in the waveform below.



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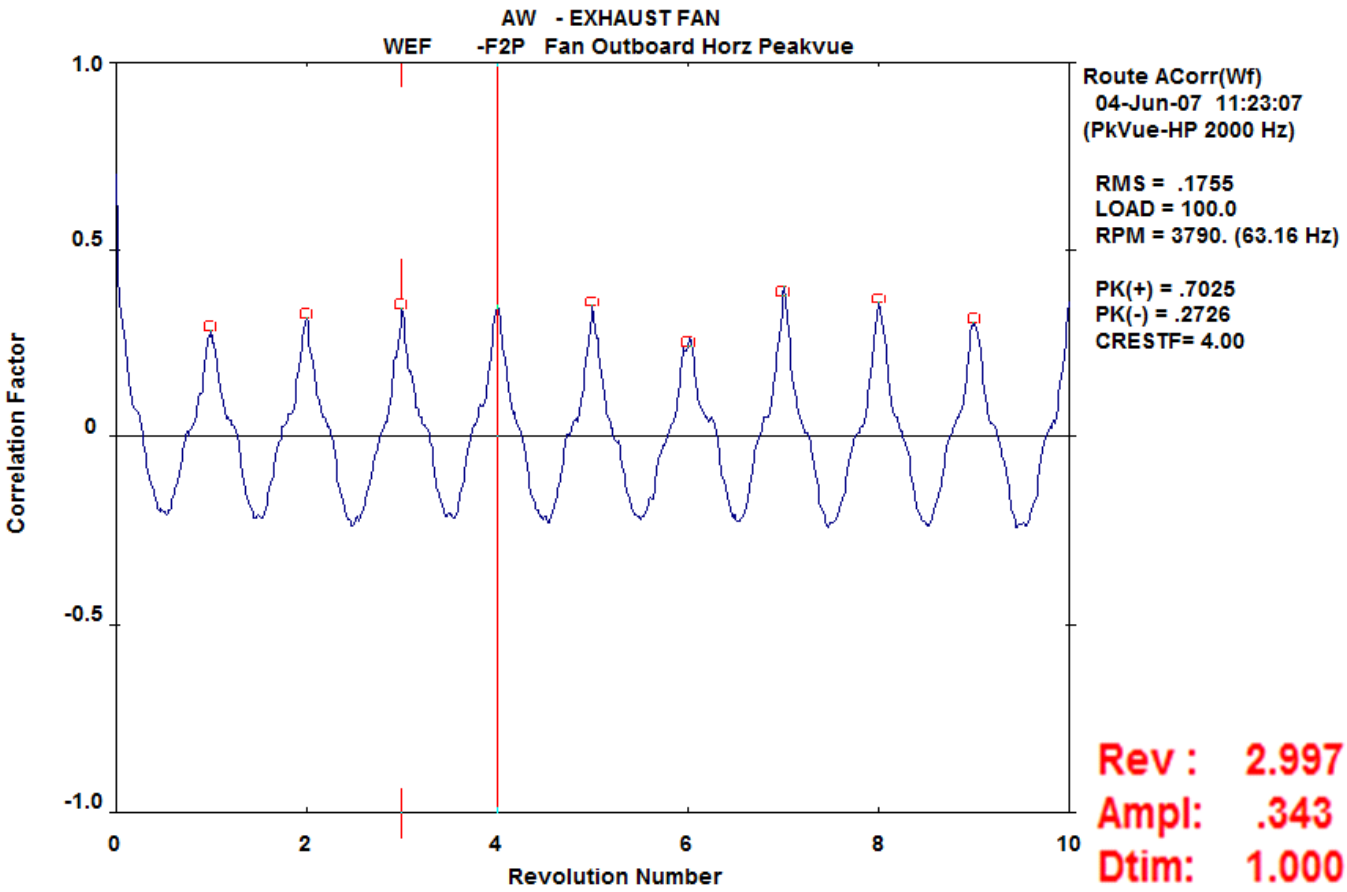
The PeakVue waveform autocorrelation is relatively low with very wide-skirted peaks. Both characteristics suggest a non-periodic signal. In this example, the fan bearings were found to be very loose on the shaft.



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Case History #3 - Phase 1 Stack Fan

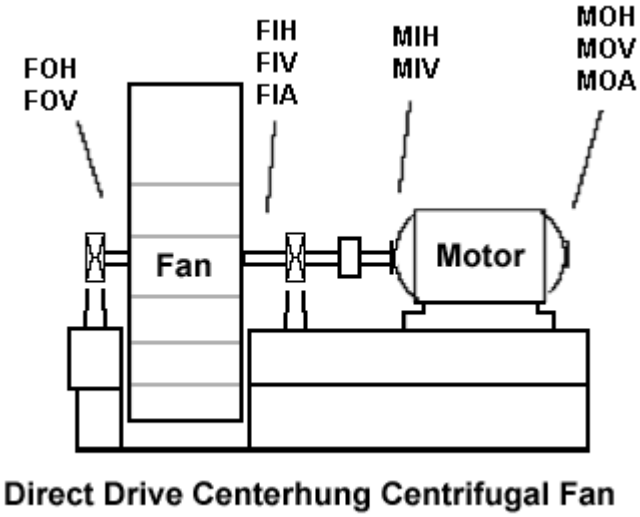

Figure 12

◆ 250 Horsepower Motor
◆ Center Hung Fan
◆ Pillowblock Bearings on Fan

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Looseness can be directional. When mounting bolts become loose on a horizontal (vertical mounting bolts) mounted machine, the looseness will appear more in the vertical direction. The data in Figure 13 is from the Fan Inboard Vertical position, a typical spectrum of a machine with structural looseness.

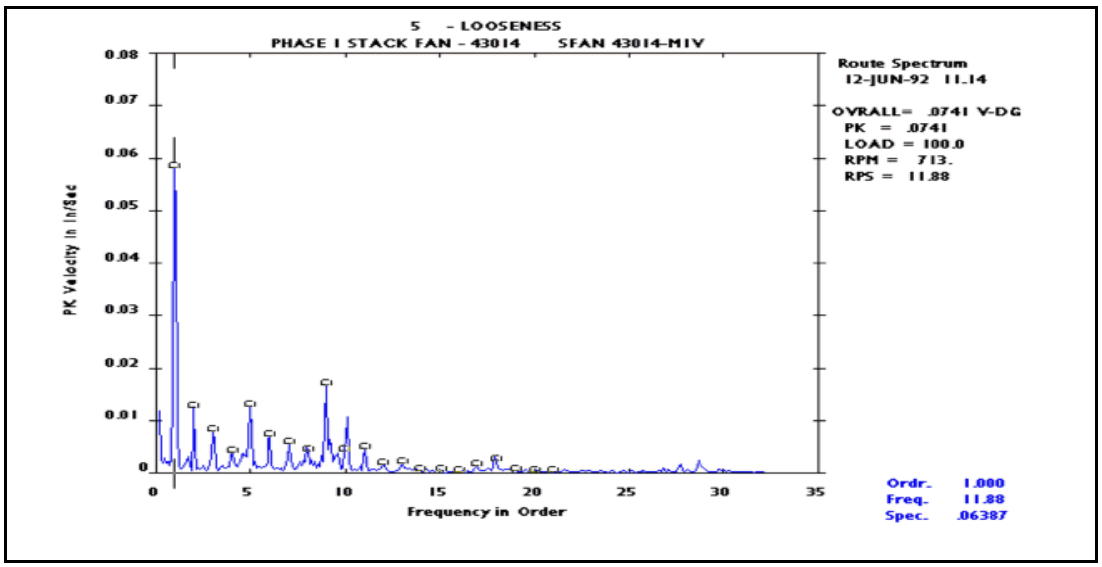

Figure 13

This Time Waveform is also representative of a looseness pattern. There is a non-repetitive type pattern. Note the relatively low G swing in the Waveform. This is also very typical of a looseness problem.

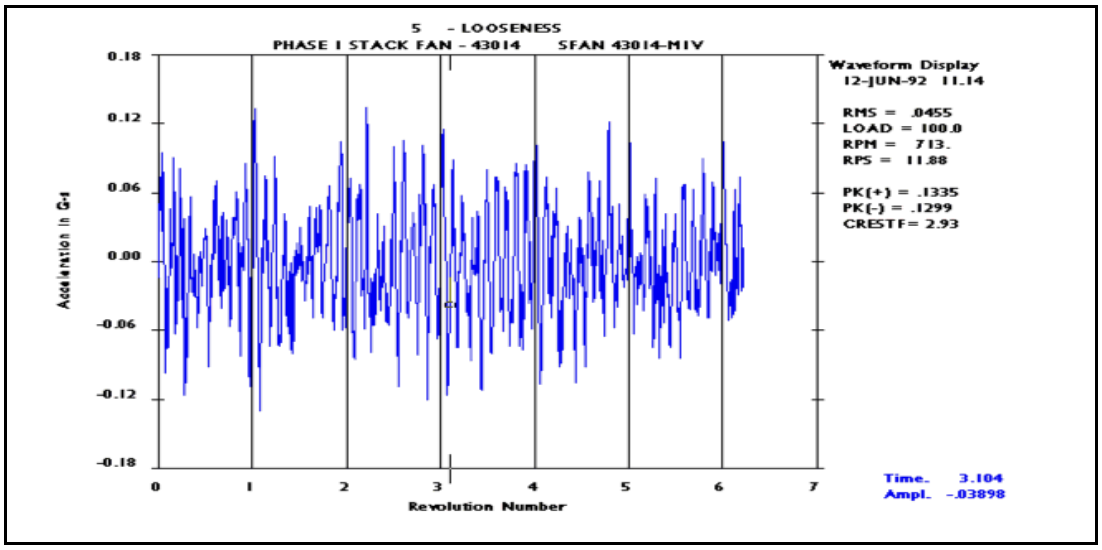

Figure 14

Phase data on this machine would reveal very erratic or unstable data.

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Review Questions

1. Name two types of looseness.

2. Describe the spectral pattern most associated with looseness.

3. Describe the waveform pattern most associated with looseness.

4. How would an analyst identify the exact amount of excessive clearance in a bearing?

5. Describe the phase characteristic most commonly associated with looseness.

6. The appearance of fractional harmonics of turning speed in the spectrum indicate what?

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Resonance

Section 11

Objectives

- Define resonance and describe its effect on machinery.
- Understand three parameters that effect resonance.
- Identify single-channel tools and their limitations.
- Determine various methods of identifying resonance.
- Explain methods to measure natural frequencies.

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Resonance

Resonance is defined as a condition that occurs when a forcing function excites a natural frequency. When resonance occurs, the vibration response to the forcing function is often amplified greatly. The vibration amplitude may be very high, although the force which is causing the vibration may be minimal. To better describe what resonance is, the term **natural frequency** must be understood.

Natural frequencies are inherent in all machines, machine components and structures. A natural frequency is a frequency which, when excited, results in the amplification of the response of the force. Excited natural frequencies also tend to continue to vibrate even after the forcing function is removed. Consider a bell as an example. When a bell is struck, amplification occurs at its natural frequency, which causes it to emit a specific audible tone. The tone the bell makes corresponds to the bell's natural frequency. Also, note that the bell will continue to produce that tone long after the impact.

Unfortunately, most machinery natural frequencies are not measured or known. Commonly, the high amplitude of vibration caused by resonance is mistaken for a high level of vibration force, which then leads to unnecessary and ineffective machinery repairs.

Identifying Resonance

3:1 Rule

One of the most common results of resonance is an abnormally high vibration amplitude as measured in only one direction. This is known as the 3-to-1 rule. This rule states that if the vibration amplitude at a single measurement location is three times higher in one direction than it is in another direction, resonance is likely. The theory behind this rule is logical. In rotating equipment, the forces from imbalance and misalignment are the same in each radial direction. For instance, the horizontal force from imbalance is the same as the vertical force. So if the horizontal and vertical forces are the same, it just makes sense that the vibration amplitudes in those directions should also be the same. There may be differences in amplitude because of differences in directional stiffness, but that difference should not result in an amplitude that is more than three times higher in one direction than it is in the other.

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Variable Speed Equipment

Another sign of resonance that is often identified on variable speed equipment is an increase in vibration amplitude at specific speed ranges. The analyst may notice that the vibration amplitude is always significantly higher at a certain equipment RPM, but when the RPM is either below or above the specified range, the vibration amplitude is significantly lower. In this situation, there is a very good possibility that the machine is resonating while running at the RPM in question.

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Measuring Natural Frequencies

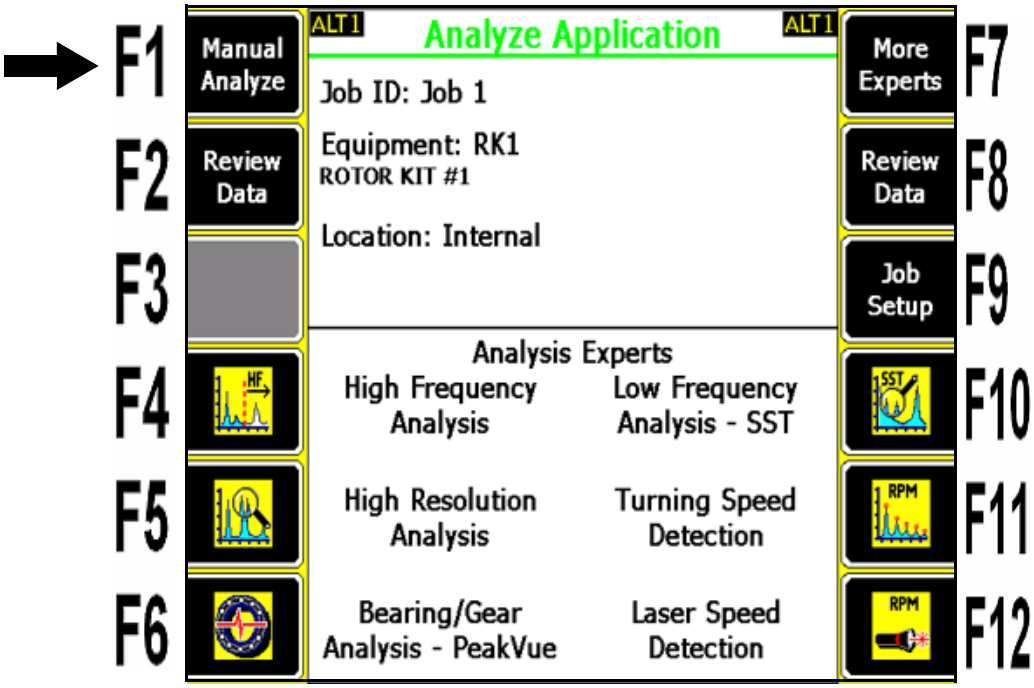
The measurement of natural frequencies on rotating equipment and components can be done many different ways. The different methods of testing are broken down into two categories:

- Coast Down (or Ramp Up)
- Impact

Peak Hold Coast Down

One method of coast down testing, Monitor Peak and Phase, was covered in a previous section of this manual. Another method for performing a coast down test is known as the **Peak Hold** method. While the Monitor Peak and Phase method is preferred, the Peak Hold method is easier to perform and does not require the use of a photo-tach or phase measurement device.

Starting at the analyzer's **Analyze Application** screen, begin the process of manual data acquisition by pressing **F1 - Manual Analyze**.



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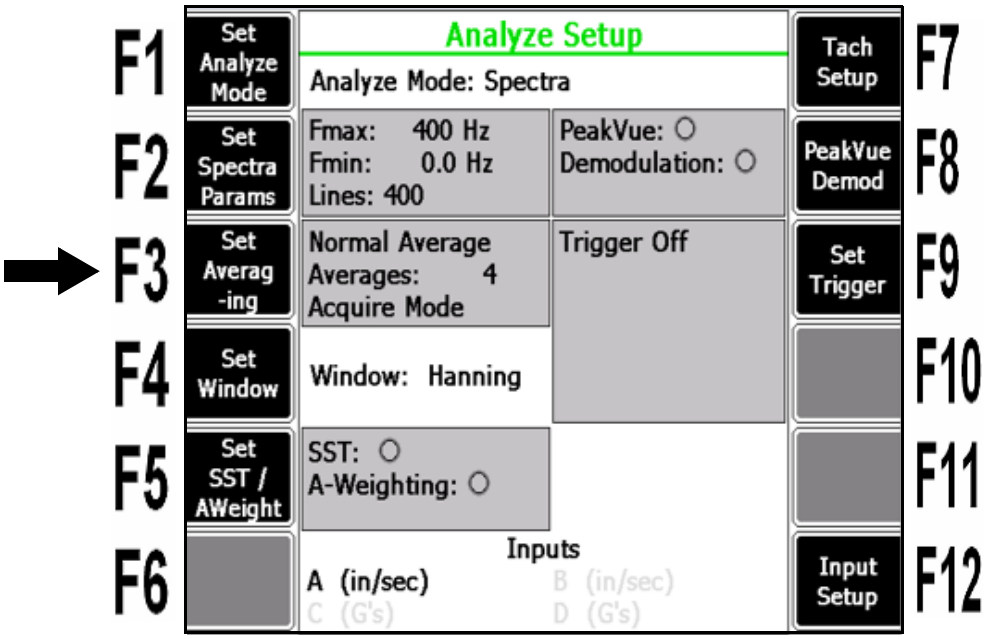
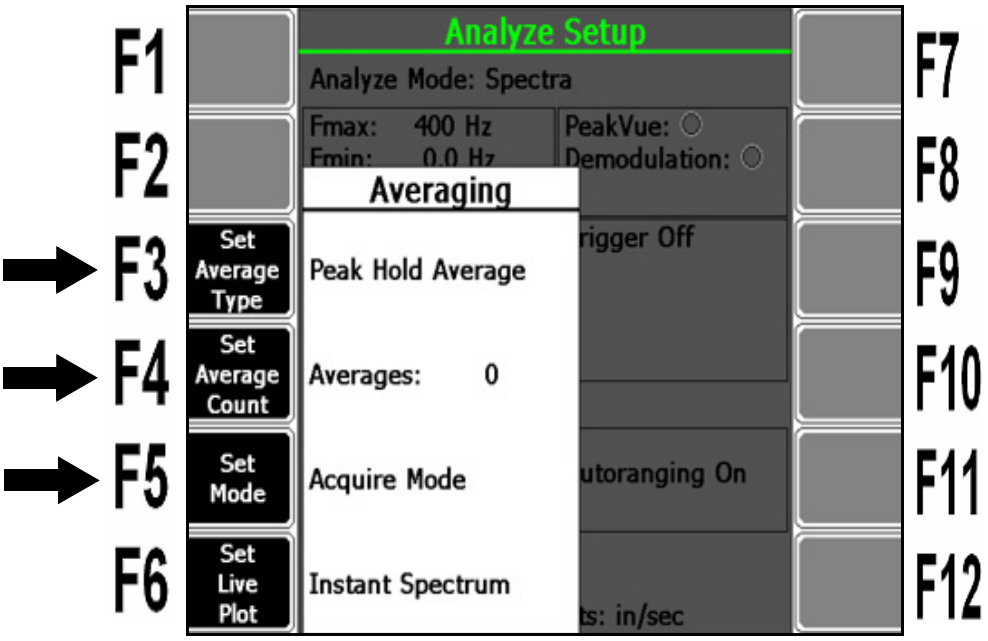
Make sure the analyzer mode is set to **Spectra**, and set the spectra parameters as shown in the example below. Press **F3 - Set Averaging**.



Set Average Type to **Peak Hold Average**. Set Average Count to **0**, and ensure the Mode is set to **Acquire Mode**.



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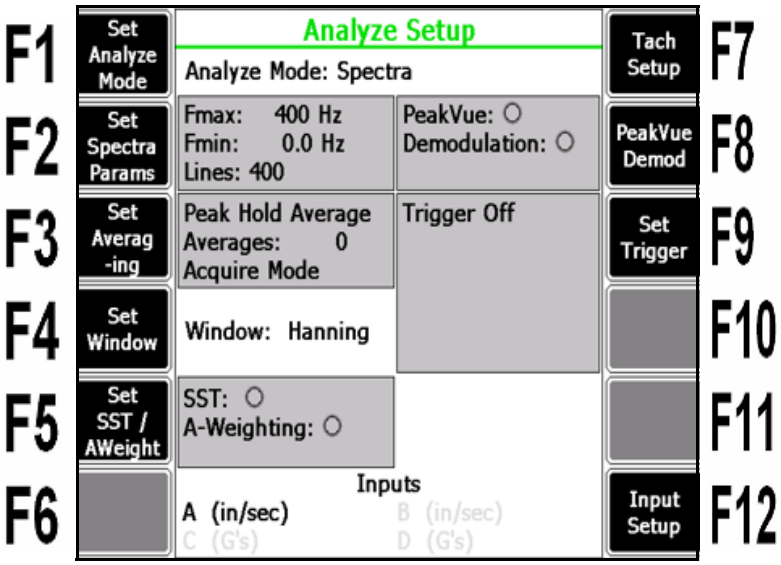
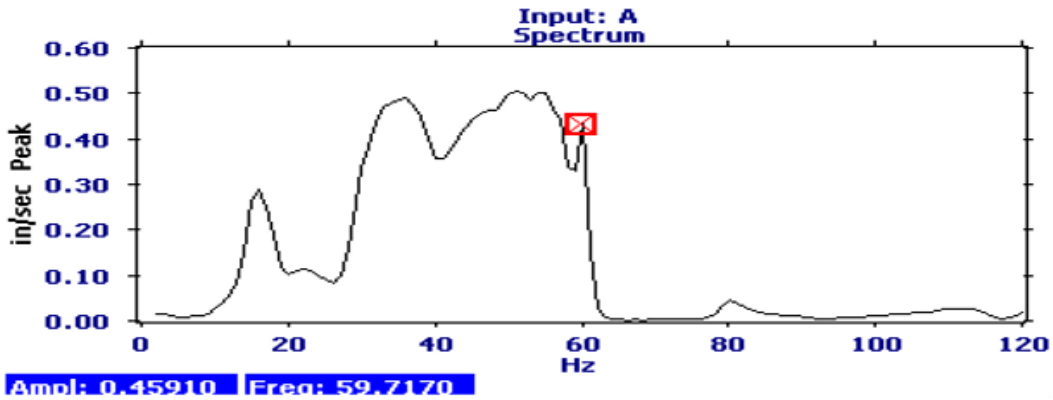
Attach the sensor to the rotor kit and, with the machine running full speed, press **Enter** to begin data acquisition. When the data appears on the screen, power off the rotor kit while continuing to acquire data. Once the rotor coasts to a complete stop, press the **Stop** on the analyzer.



In this example, the peak at 59.7 Hz is the 1 x RPM peak, while the machine is running full speed. It is assumed that the force causing this vibration at 1 x RPM is imbalance. It is also assumed that the vibration energy to the left of that peak in the spectrum is the 1 x RPM amplitude as the machine coasts down. Based on those assumptions, the vibration force is consistently less as the RPM is reduced. Therefore, the vibration amplitude should also be consistently lower. If the vibration amplitude increases while the vibration force decreases, amplification must be occurring. In this Peak Hold spectrum, any peaks to the left of the peak at 59.7 Hz represent natural frequencies.



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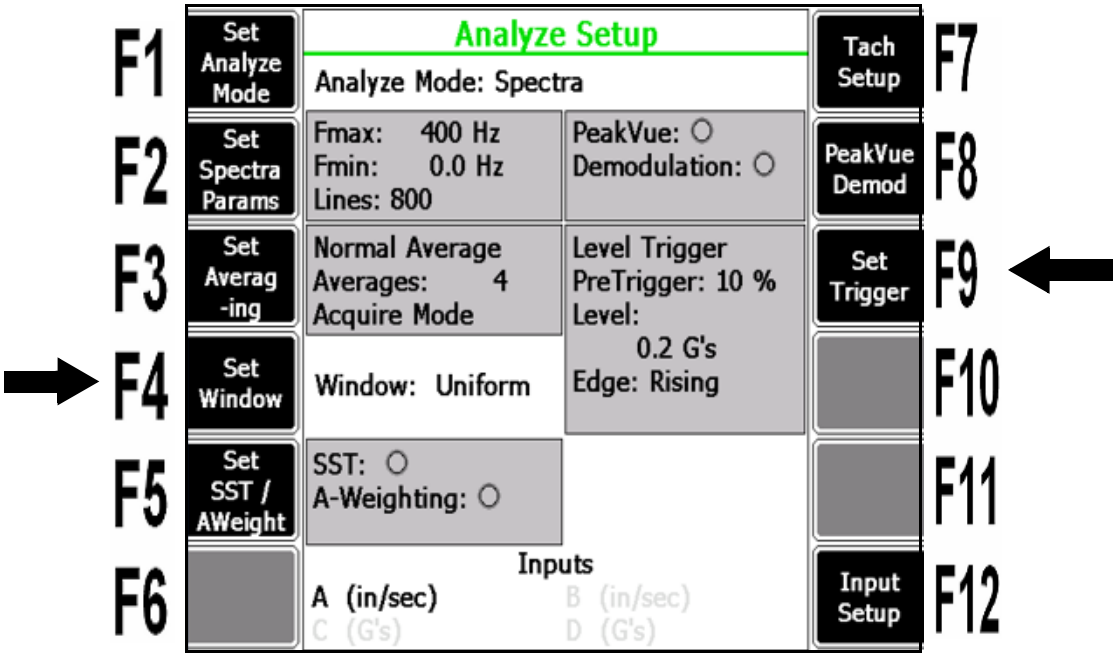
Impact Test

An **impact** test is another method for identifying natural frequencies. In an impact test, the machine, component or structure is impacted only once (per average). The purpose of the impact is to generate completely random vibration energy. In the spectrum, that random vibration energy should result in a significant amount of broadband energy. Any peaks in that broadband energy should be a result of amplification due to resonance. The peaks in the impact spectrum represent natural frequencies.



The typical acquisition set up for a single channel impact test is shown above. Note, there are a few differences between this and normal route data collection. First, the window type has been changed from Hanning to **Uniform**. The window is changed so the impact waveform will not be affected, allowing the true impact shape to be captured. Another difference is the use of a **Level Trigger**. The Level Trigger is used so that the analyzer will begin the data capture precisely when the impact occurs. Also, a **Pre-Trigger** of 10% is applied, so that a certain amount of data before the impact will appear in the waveform.

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Class Exercise

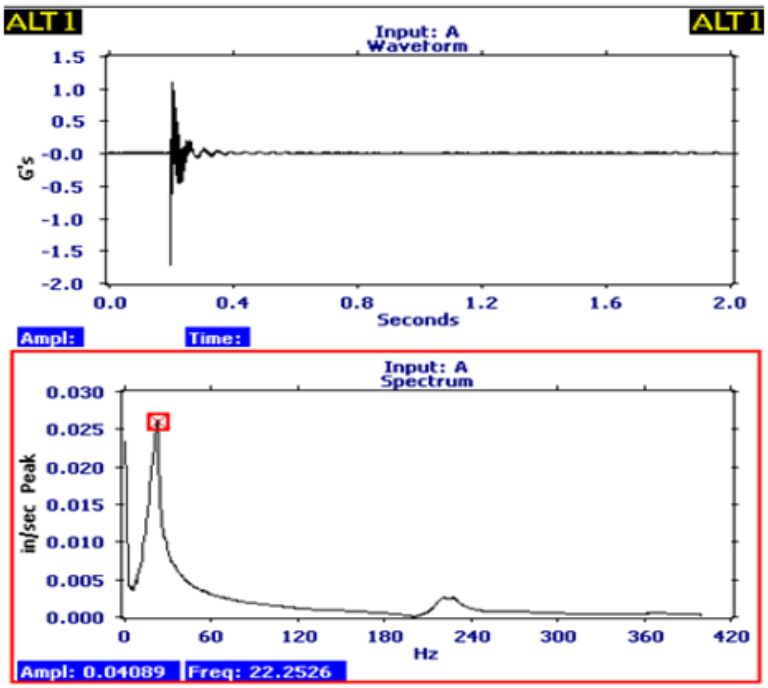
Use these settings to perform an impact test on the class rotor kit. Place one sensor in the vertical direction on the machine, and press **Enter**. When the analyzer screen says **Waiting for Trigger**, impact the machine vertically one time near the sensor. Try to make the impact cleanly, making sure to lift the hammer quickly. The motion should be more like ringing a bell rather than driving a nail.

When the message **Waiting for Trigger** appears on the screen again, impact the machine again, trying to use the same amount of force as the first impact. Repeat the impact until all four averages have been acquired.

Note

Preferred hammers include rawhide, rubber or nylon tipped. Avoid dead-blow and steel faced hammers. The composition of the dead-blow hammer heads may compromise the success of the test. Steel hammer heads tend to bounce as they land generating multiple impacts in the average. A length of 4x4 lumber may also be used.

The data screen should include the last acquired waveform along with the averaged spectrum. Analyze the spectrum for the presence of any obvious peaks. In this example, there is a suspected natural frequency at 22.25 Hz.



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Repeat the process by moving the sensor to the horizontal direction and impacting the machine in the horizontal direction. Analyze the results and compare the results with the vertical impact data.

Which Type of Test is Best?

Impact and coast down tests can both be used to identify natural frequencies. The decision about when to use one type of test instead of another is often made based on a number of factors. For instance, coast down tests can only be performed on rotating equipment. If an analyst needs to know the natural frequency of a structure, or a non-rotating piece of equipment, the impact method is almost the only option available. If the analyst needs to know the natural frequencies of a large turbine generator rotor, a coast down test is often preferred, since it would be difficult to impact the rotor with enough force to properly excite its natural frequencies. If both types of tests are possible, then both types of test should be performed to ensure that important natural frequency information is documented.

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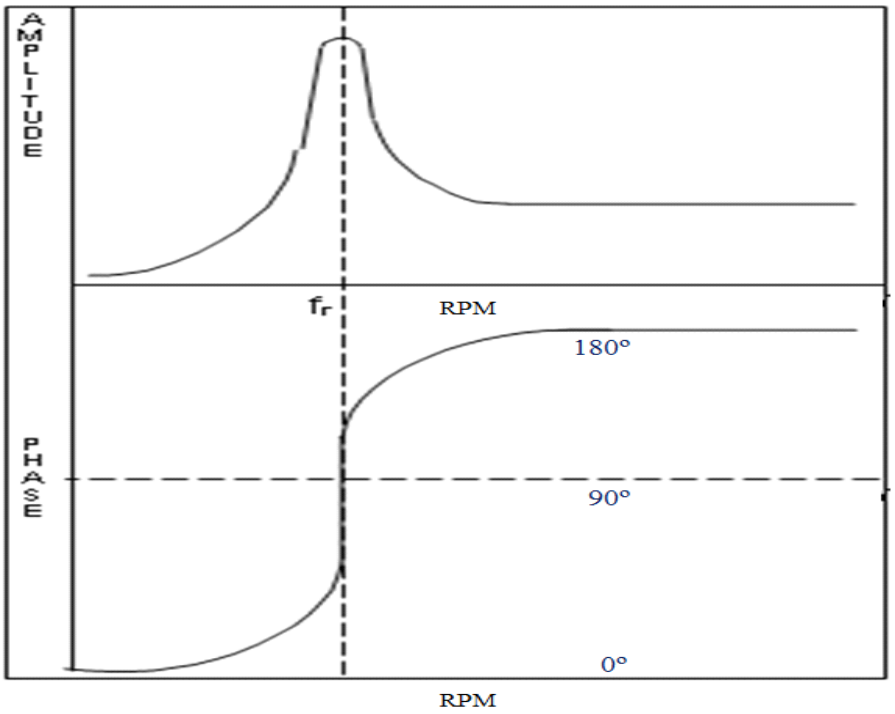
Phase and Resonance

The phase characteristics of a rotor change according to its RPM and its Critical Speed. The term **Critical Speed** is used to describe the RPM at which the rotor will excite its own natural frequency. When a rotor's RPM is well below its critical speed, it is described as a **Rigid Rotor**. A rigid rotor has a phase lag close to zero. As the rotor's RPM increases and it approaches its critical speed, it begins to resonate. When it resonates, the shaft begins to bend. This bending motion causes a time difference between where the phase is being measured, (at the reflective tape or keyway), and where the vibration is being measured, (at the bearing). This causes the phase to shift. The closer the RPM gets to the critical speed, the closer the measured phase value gets to 90 degrees. When running at its critical speed, the phase is 90 degrees. As the RPM increases further and becomes higher than the critical speed, the phase continues to increase until, finally, it reaches 180 degrees, when the RPM is well above the critical speed.



During a coast down, the process is reversed. If the rotor's RPM is well above the critical speed, the phase will be close to 180 degrees. As the rotor coasts down and nears the critical speed, the phase value decreases. At the critical speed, the amplitude peaks, and the phase is 90 degrees. As the rotor RPM decreases further, the phase continues to shift until it reaches 0 degrees.

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Mass, Stiffness and Damping

There are three physical characteristics that control the natural frequencies of any machinery component. They are **Mass, Stiffness** and **Damping**. Mass and stiffness control the actual frequency itself. Damping controls how much amplification occurs, when the natural frequency is excited.

Mass and Stiffness

To help explain how mass and stiffness control the natural frequency, consider the equation that is used to calculate the natural frequency.

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

where

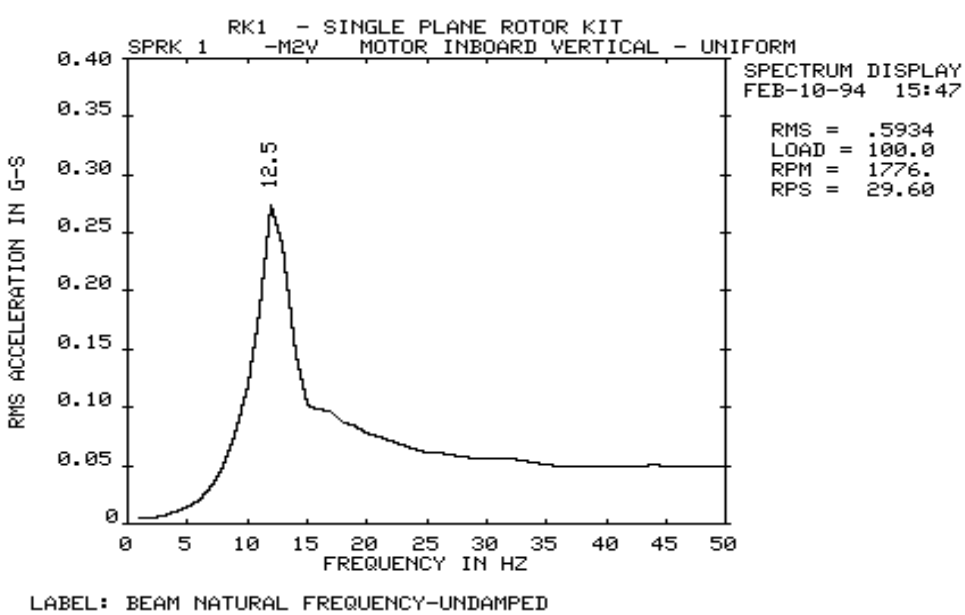
f_n = damped natural frequency (Hz)

k = stiffness (lb/in)

m = mass (lb-sec²/in)

Based on the equation, a machine with more mass and less stiffness will have a comparably lower natural frequency than a machine that has less mass and more stiffness. It is the mass and the stiffness that control what the machine's natural frequencies are.

In this example, an impact test is used to measure the machine's natural frequency of 12.5 Hz.



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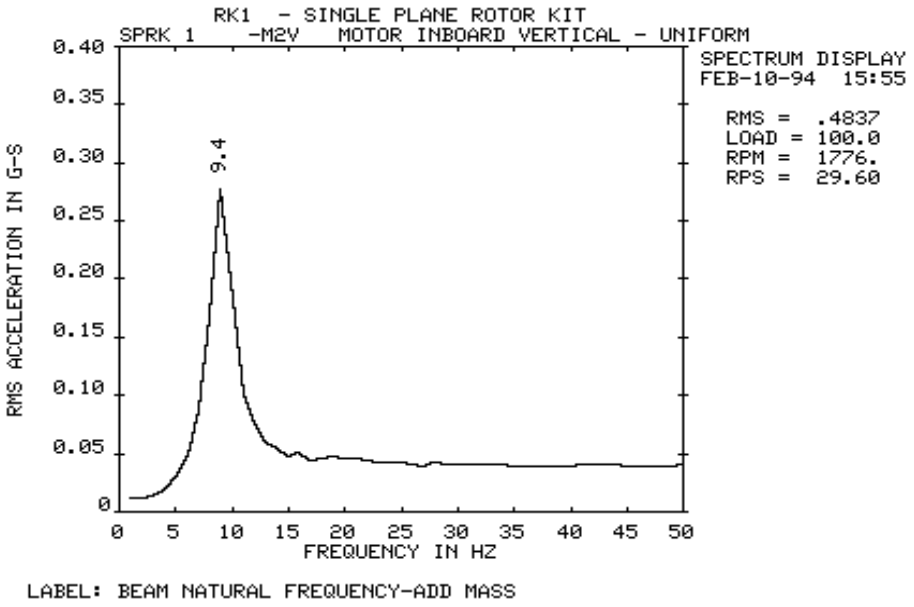
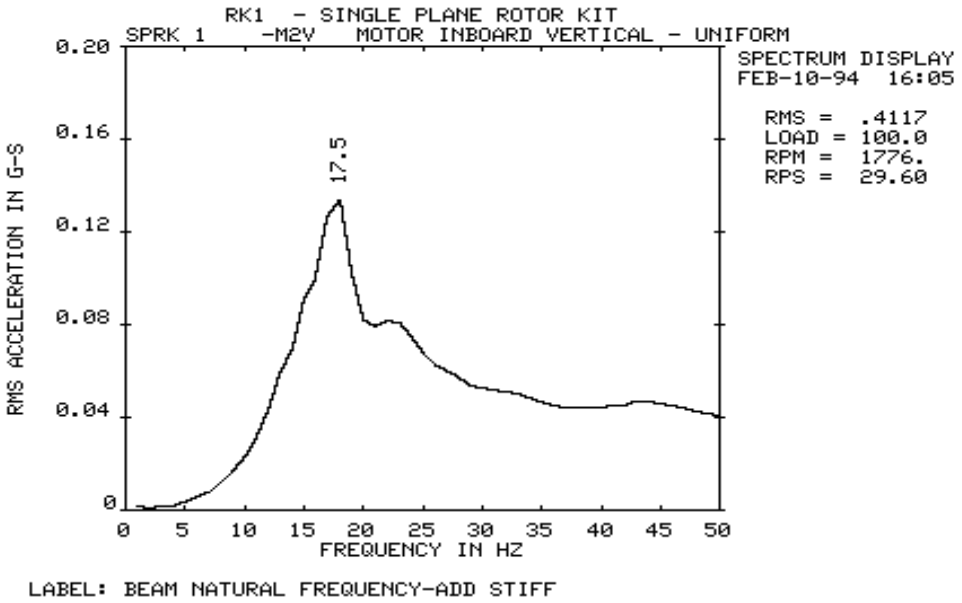
Mass is added to the machine and the impact test is performed in the same manner. The result is a lower natural frequency of 9.4 Hz.



The added mass is then removed and the machine is stiffened. The increased stiffness results in a higher natural frequency. In this case, the natural frequency has climbed to 17.5 Hz.



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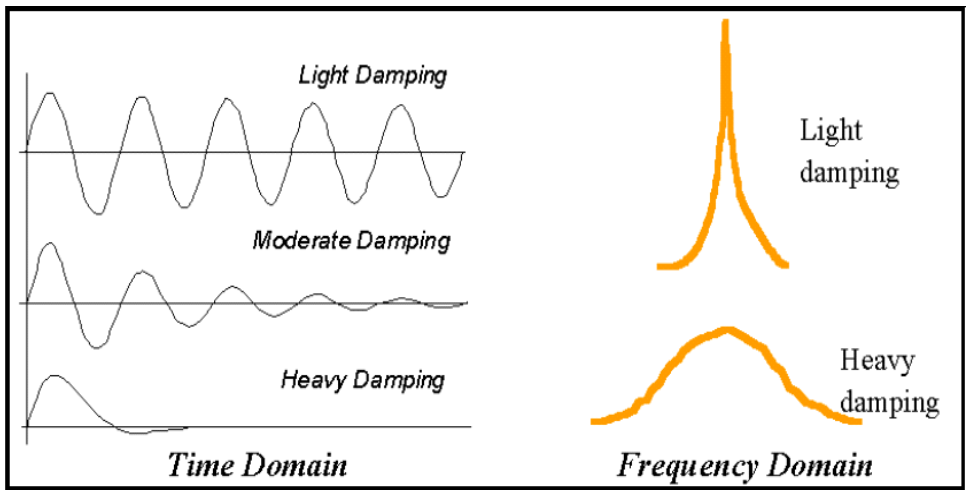
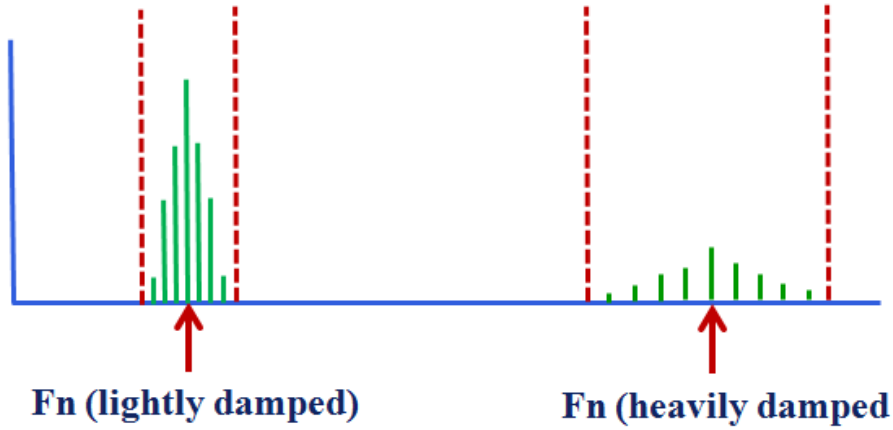
Damping

Damping is the property that determines how much amplification occurs when a natural frequency is excited. Comparatively, when a system is lightly damped, the amplification curve is taller, but it is also narrower. This means that a forcing function needs to be closer to the natural frequency itself in order to have any effect at all. When the forcing function is on the curve, the amount of amplification will be very great. In comparison, if the system is more heavily damped, the curve will be wider and lower. In this case, the forcing function may be affected even further away from the actual natural frequency, but the amplification will be less.



Because differently damped systems have different natural frequency curves, the exact separation that a forcing function needs from a natural frequency is not the same for every machine. Most machines in industry are lightly damped. **The rule of thumb that applies to most machines is that any forcing function of the machine should be at least 20 percent away from any natural frequency**.



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Options for Correcting a Resonance Problem

All structures and machines have natural frequencies, but resonance will not occur unless there are forcing functions present that excite those natural frequencies. If resonance is found to be the problem, there are three options to consider.

1. Move the forcing function frequency away from the natural frequency. This involves either changing the machine RPM or changing the number of vanes or blades on a pump or fan impeller.
2. Move the natural frequency away from the forcing function. This involves modifying the machine, either adding or removing mass or stiffness. It should be noted that machinery modifications should only be performed by qualified mechanical engineers.
3. Add damping to reduce the amplification factor of the resonance.

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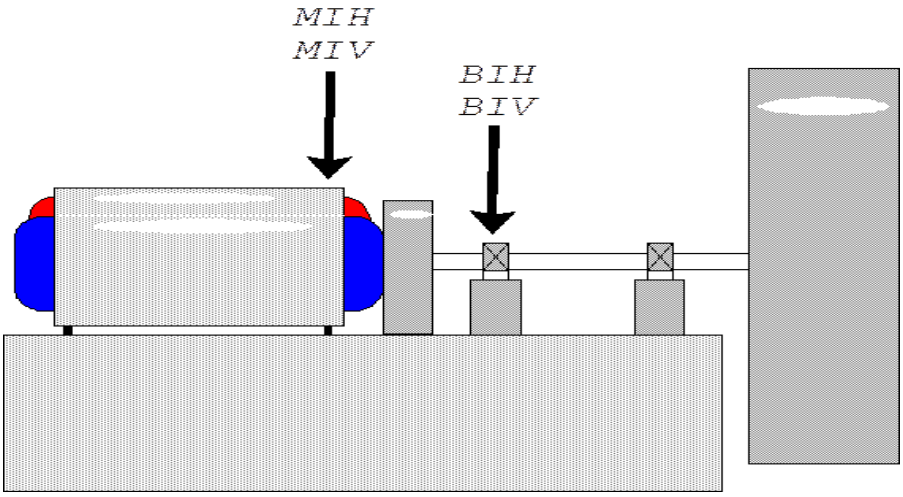
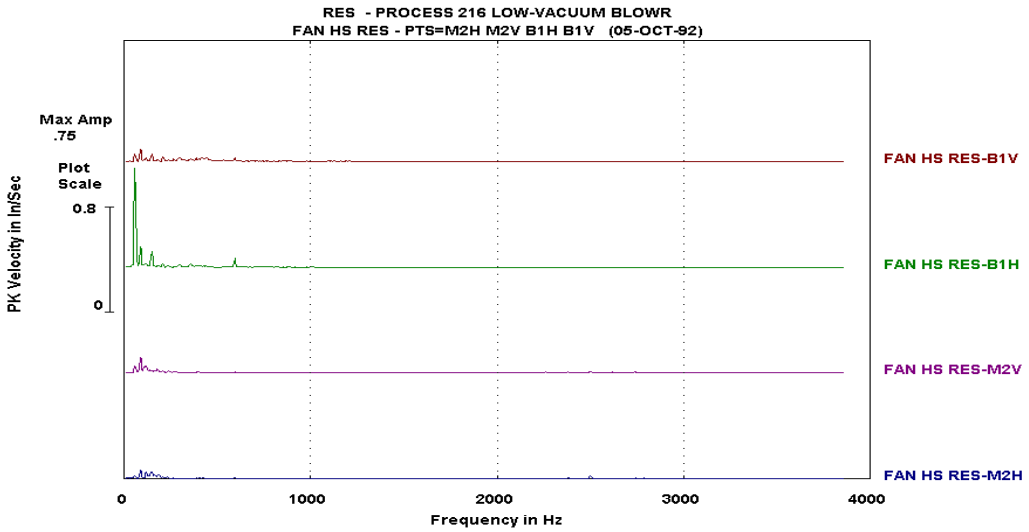
Case Histories

Resonance Case History #1

- 125 HP 1200 RPM motor
- In-line drive overhung fan



The data below is a multiple point spectrum. This data was acquired from the motor and blower inboard horizontal and vertical measurement locations. Notice that the vibration amplitude is significantly higher on the blower inboard horizontal direction than at any other position.



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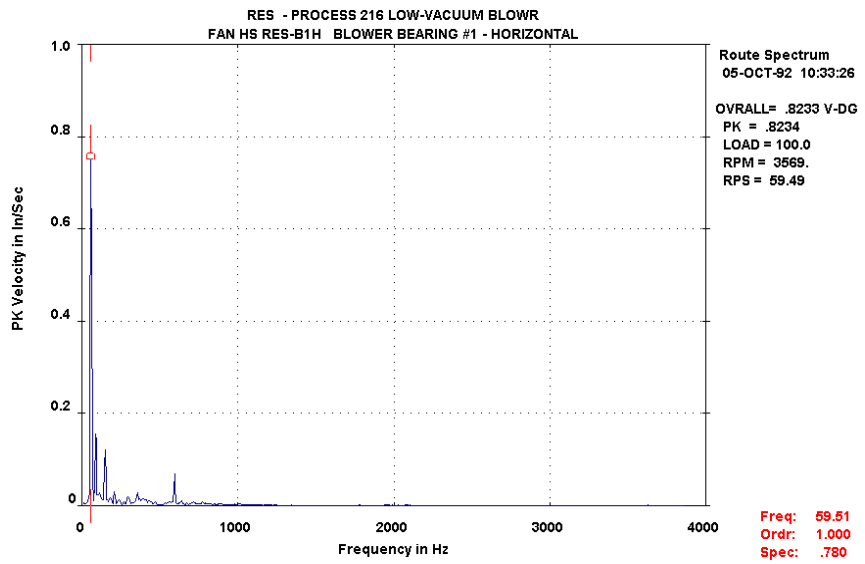
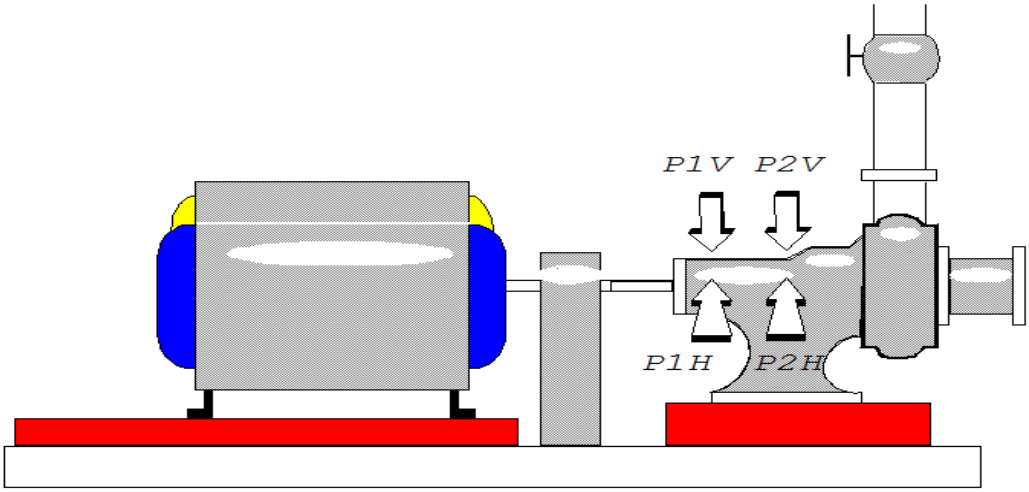
A single spectrum display of only the blower inboard horizontal point reveals that the dominant peak is 1 x RPM. The amplitude at that frequency is over .7 in/sec, which is more than 10 times higher than the vertical vibration amplitude at that same location. While it may be tempting to blame imbalance as the sole reason for the vibration, the reason that the amplitude is so high is actually resonance. If it were due to imbalance, the vertical amplitude would be much higher.



Resonance Case History #2

- 20 HP 3600 RPM motor
- In-line driven overhung pump



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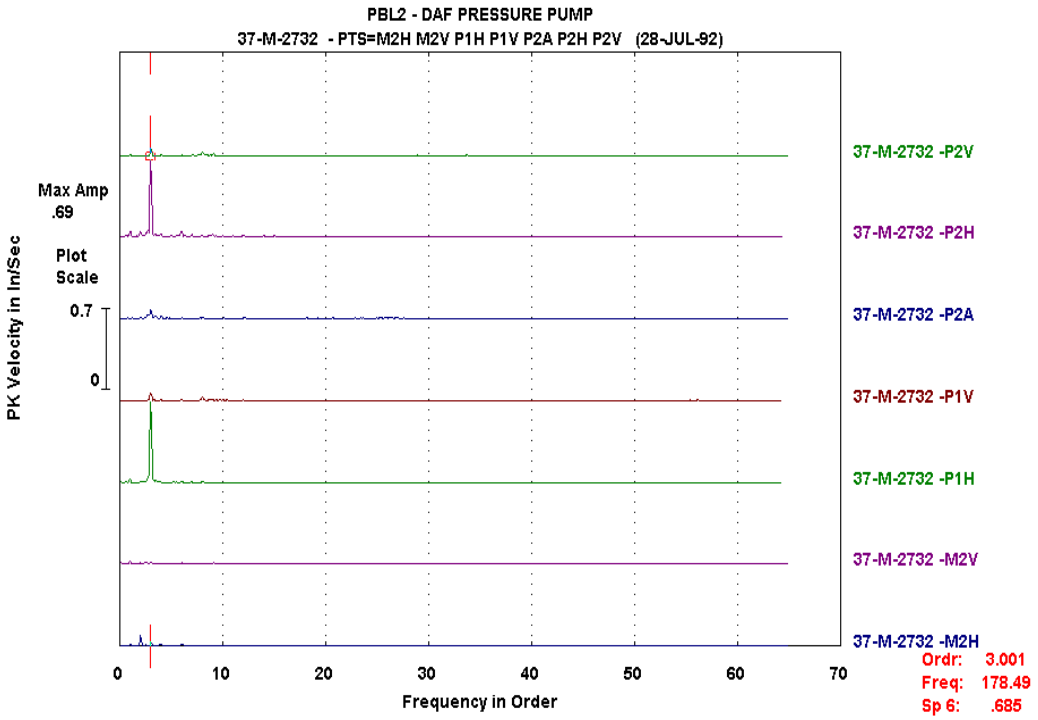
A multiple spectrum display shows that the amplitude of vibration is significantly higher on the pump in the horizontal direction than anywhere else on the machine.



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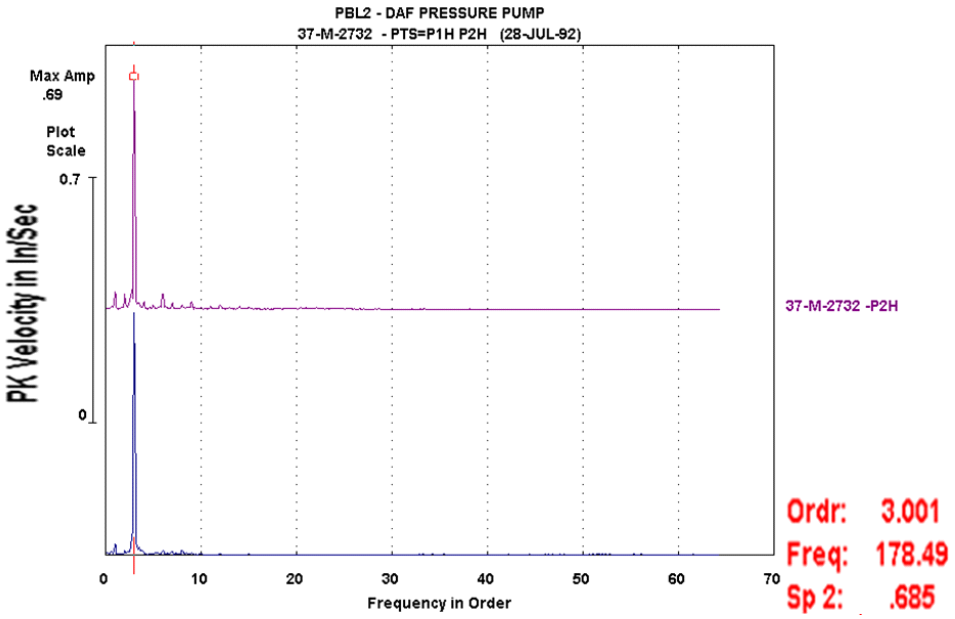
A closer analysis of the spectra indicates that the vibration frequency is 3 orders. This pump has a three vane impeller, so this is this pump's vane pass frequency. In most cases, an amplitude increase at the vane pass frequency would be a result of flow restriction in the pump. However, if this is caused by flow restriction, why would the high amplitude be present only in the horizontal direction?

Remember the 3-to-1 rule. If the ratio of vibration amplitude between the horizontal and vertical measurements on the same measurement location is more than 3:1, consider resonance as a possible cause.



An impact test was performed on the pump and a natural frequency was identified at 173 Hz, which is almost exactly this pumps vane pass frequency.

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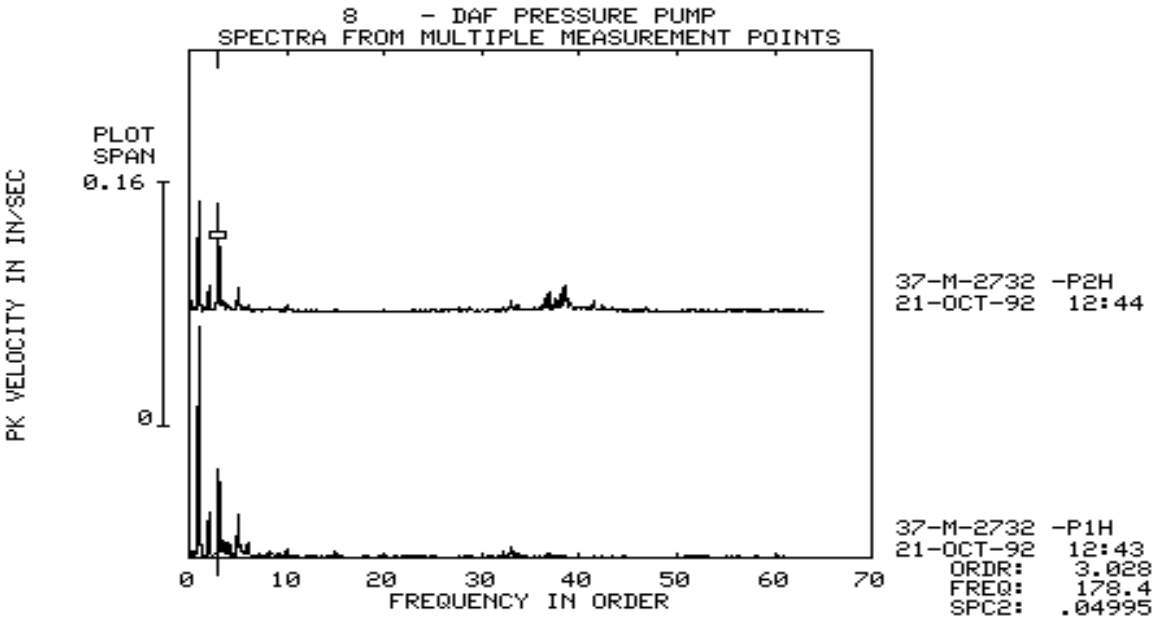
The pump's three vane impeller was replaced with a four vane impeller. This single modification resulted in a significant decrease in vibration amplitude. The amplitude at 3 x RPM is now less than .05 in/sec, compared to nearly .7 in/sec before the modification.



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Review Questions

1. What is a Critical Speed?

2. Describe two modifications that can be performed to increase a machine's natural frequency.

3. True or False? A dead-blow hammer is a good choice for performing an impact test.

4. Name at least two methods of collecting data on a machine during coast down.

5. True or False? The Hanning window is used to perform an impact test.

6. What is a rotor's phase lag when it is running at its critical speed?

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Electrical Defects

Section 12

Objectives

- Explain the electrical defects associated with AC induction motors.
- Describe the function of Induction motors.
- Identify the various electrical defects associated with induction motors.
- Explain analysis techniques using single-channel data acquisition.

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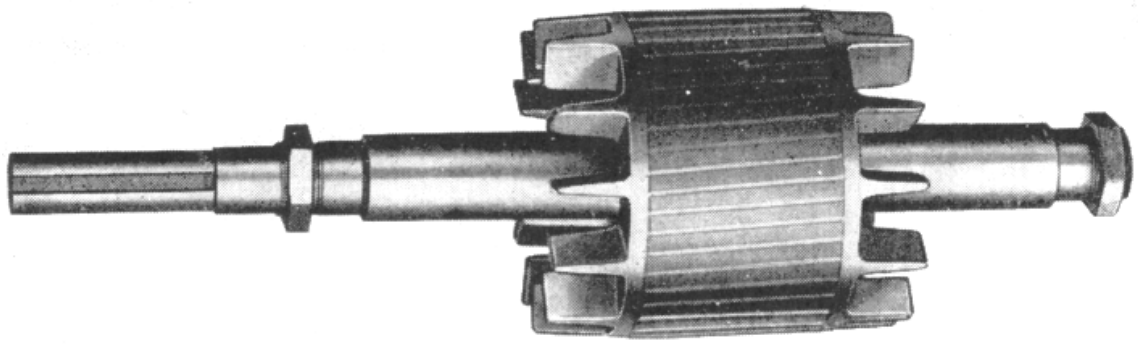
General Description

The majority of mechanical process equipment in industry is driven by electric motors.

While DC motors are still used in a number of industries, they are being phased out by variable speed AC induction motors.

The most common type of AC motor is the induction motor. This type of motor has two main components, the rotor and the stator. This section primarily focuses on motors which use a squirrel cage type rotor, which are the most typical.

The Rotor



The rotor is the rotational element of the electric motor. A squirrel cage rotor design is used in most induction motors. This design uses a laminated slotted core in which the conductive material for the rotor bars is placed in the slots. The rotor bars are then shorted together by the end rings. Any manufacturing deficiencies will be detectable using the data collection techniques discussed later.

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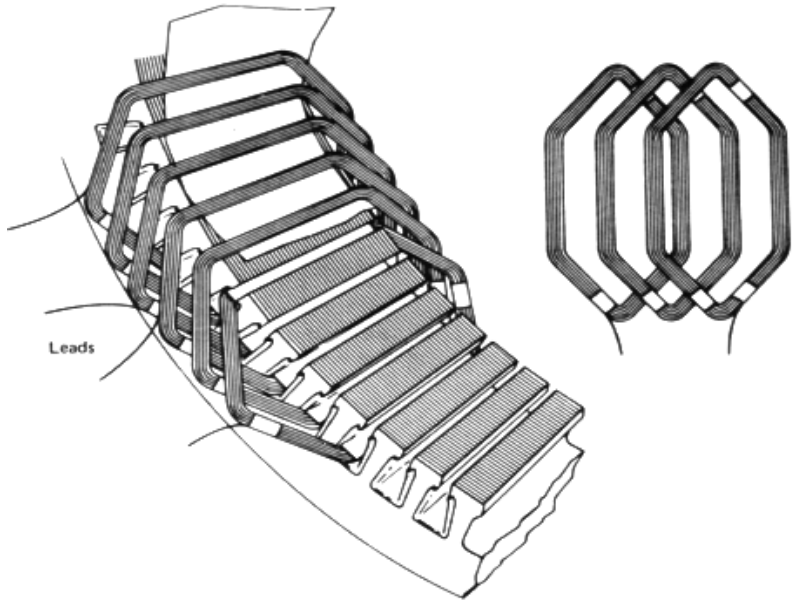
The Stator

The stator is the stationary part of the induction motor that consists of a cylindrical laminated slotted core and the windings that are placed in the frame.

The stator windings are comprised of turns of copper wire that form a coil. These coils are placed in the slots of the core. The windings are separated from each other and the core by insulation. The figure below shows how the individual coils are placed in the stator and how the coils are connected to form a pole.



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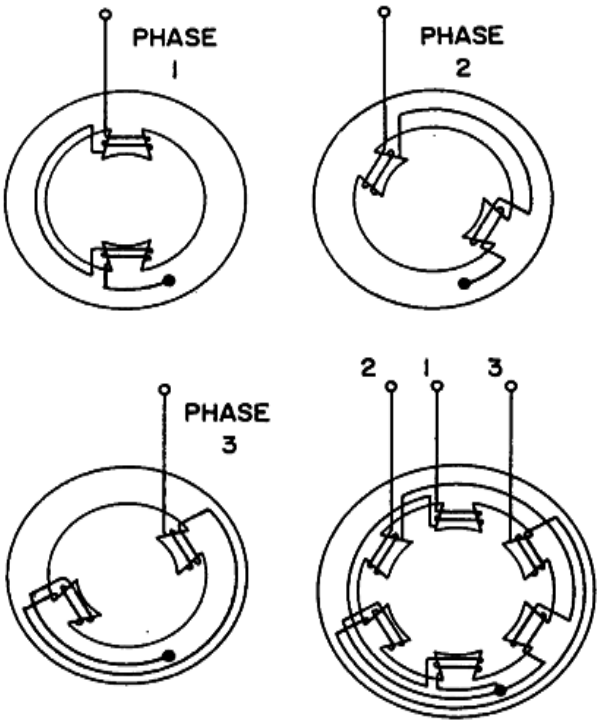
A simplified three phase, 2-pole winding diagram is displayed here. Notice how a winding from each phase makes up the motor's two poles.



When the three phase stator of an induction motor is connected to a three phase source, three independent currents flow. One current flows in each of the three phase windings. This produces a rotating magnetic field in the stator of the motor which rotates at a speed given by the following formula.

Synchronous RPM = 7200 ÷ #poles

Note

7200 is 2 x electrical line frequency in North America.

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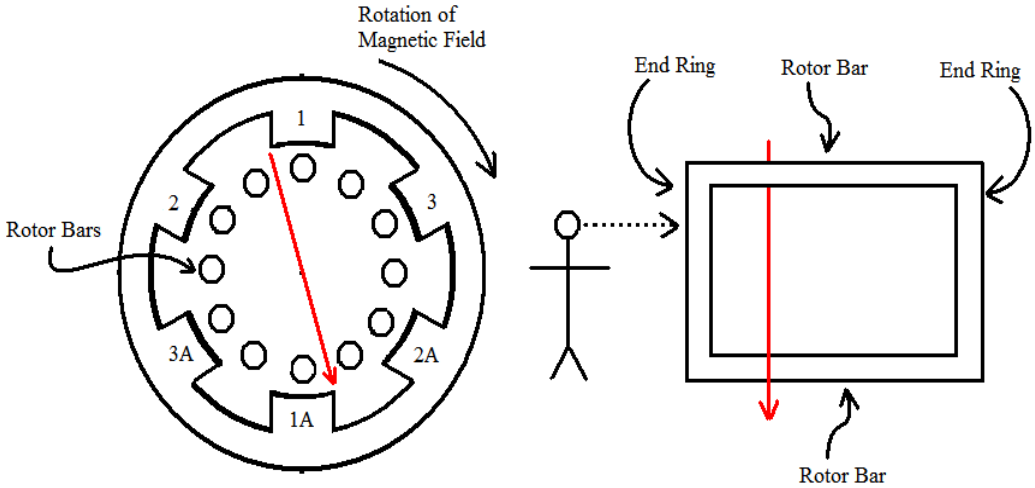
Theory of Operation

An AC induction motor consists of a rotor placed in a three phase stator. The purpose of an AC induction electric motor is to convert electrical energy into mechanical energy. The actual conversion of energy takes place in the rotor. This conversion of power is transferred across the air gap inductively. The relative motion between the revolving field and the rotor causes the rotor bars to be cut by the lines of flux, which generates a rotor voltage causing current to flow in the rotor. The rotor now rotates from the developed torque. The difference between the RPM of the revolving field and the RPM of the rotor is called the **slip**.

Consider the diagram below; initially the motor is not turning. The three phase stator has power applied to it causing a rotating magnetic field in the stator. The magnetic field is rotating clockwise and since it is a two pole motor, it rotates at 3600 RPM. In the diagram, the magnetic field has been frozen just before it cuts the rotor bars under phase 1, coils 1 and 1A. As the magnetic field sweeps through the rotor bars, it causes a very large current to flow in the rotor bars and end rings.



The current in the rotor bars has a magnetic field associated with it. This magnetic field is at 90 degrees to the stator magnetic field. These two magnetic fields interact. The north pole of the stator pushes on the north pole of the rotor magnetic field. The north pole of the rotor magnetic field is attracted to the south pole of the stator magnetic field. The south pole of the stator magnetic field pushes on the south pole of the rotor magnetic field. And the south pole of the rotor magnetic field is attracted to the north pole of the stator magnetic field.

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All this pushing and pulling causes torque in the rotor, and the rotor accelerates the load to nearly the speed of the stator. The rotor cannot reach synchronous speed. If it reached synchronous speed, then there would be no difference in speed between the rotor bars and the magnetic field in the stator.

It is the magnetic lines of flux from the stator cutting the rotor bars that gives the rotor a magnetic field. The larger the difference in speed between the stator magnetic field and the rotor, the greater the current flow in the rotor bars and the stronger the magnetic field in the rotor. The speed of the rotor stabilizes at a point where the magnetic field in the rotor is sufficient to develop enough torque to turn the load.

Calculating Defect Frequencies

Number of Poles

To calculate the number of poles a motor has, divide 2 x LF (in CPM) by the motor's rated turning speed, and then round to the nearest whole number.

 # Poles = 7200 ÷ rated RPM

Example: motor rated rpm = 1780

poles = 7200 ÷ 1780

poles = 4.04 (round down to nearest whole number)

poles = 4

Motor Synchronous (Stator) Speed

To calculate the synchronous speed of an AC induction motor, divide 2 x LF (in CPM) by the number of poles.

Synchronous Speed = 7200 ÷ # poles

Example: motor running 3560 RPM

Synchronous Speed = 7200 ÷ # poles

Synchronous Speed = 7200 ÷ 2

Synchronous Speed = 3600 CPM

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Slip Frequency

The slip frequency is the motor's synchronous speed (or stator speed) minus the motor's actual RPM (rotor speed).

Slip Frequency = Synch Speed – RPM (actual)

Example: motor running 3560 RPM

Slip Frequency = Synch Speed - RPM

Slip Frequency = 3600 - 3560

Slip Frequency = 40 CPM

Pole Pass Frequency

The pole pass frequency is the number of poles multiplied by the slip frequency.

Pole Pass Frequency = # Poles x Slip

Example: motor running 3560 RPM

Pole Pass Frequency = # Poles x Slip

Pole Pass Frequency = 2 x 40

Pole Pass Frequency = 80 CPM

Slot Pass Frequency

Defined as the frequency at which one spot in the rotor passes all of the slots in the stator assembly.

Slot Pass Frequency = # Stator Slots x Rotor RPM

Example: motor with 42 stator slots running 1743 RPM

Slot Pass Frequency = 42 x 1743

Slot Pass Frequency = 73,206 CPM or 1220.1 Hz

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Rotor Bar Pass Frequency

Defined as the frequency at which rotor bars pass a single spot in the stator assembly.

Rotor Bar Pass Frequency = # Rotor Bars x RPM

Example: motor with 44 rotor bars running 1775 RPM

Rotor Bar Pass Frequency = 44 x 1775

Rotor Bar Pass Frequency = 78,100 CPM or 1,301.7 Hz

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AC Induction Motor Analysis

Vibration analysis on electric motors can be challenging. The spectral resolution required for analysis of some electrically related problems is not typically included in normal route data collection. Also, many different electrical defects cause very similar vibrations, so identifying the exact source of the problem can be nearly impossible. In many cases, once the vibration analyst has identified a potential electrical problem in the route data, further testing is required to identify the true source of the problem. Additional testing may include high resolution spectrum analysis, motor current analysis or even offline testing with dedicated electrical testing equipment.

The first step is often an attempt to verify whether the vibration is mechanical or electrical in nature. A simple, but effective, way to do this is to monitor the vibration spectrum live, and then secure the power to the motor. If the vibration energy in question immediately disappears, the source of the vibration is electrical. If the vibration energy slowly diminishes with the turning speed of the rotor, the source of the vibration is mechanical.

Generally speaking, electrical defects can be classified into two categories: rotor problems and stator problems. Rotor problems tend to cause excessive vibration at 1 x RPM of the rotor, while stator problems tend to cause excessive vibration at 2 x Line Frequency or in some way related to 2 x Line Frequency. That is about as specific as vibration analysis can be in many cases. Once the problem has been determined to be electrical in nature, further testing needs to be done to determine the exact source of (and severity of) the problem.

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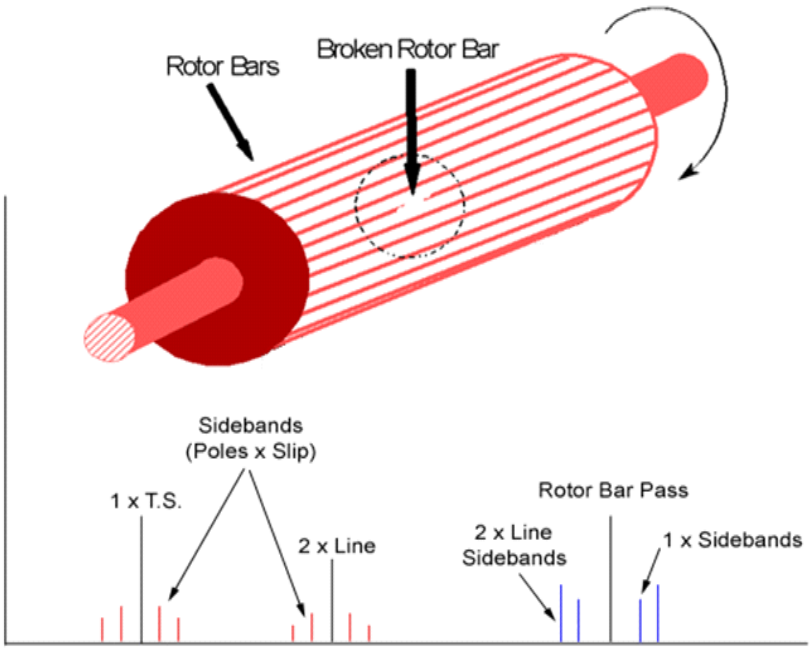
Rotor Related Defects

Broken or Open Rotor Bars



Spectrum

The dominant vibration appears at 1 x TS with sidebands spaced at a frequency equal to the Pole Passing Frequency (#poles x slip). The 1 x TS vibration will increase after startup as the rotor heats up in the bars adjacent to the broken bars. This causes an imbalance condition that worsens over time. Also, look for a 2 x line frequency spectral line in the spectrum with the rotor slot passing frequency present. This high frequency forcing function may be modulated by 1 x TS and 2 x line frequency causing sidebands at both of those frequencies. Acquire high resolution, low Fmax data to identify pole passing frequency sidebands around the peak at 1 x TS. Rotor slot frequency may also be present with sidebands spaced at 2 x line frequency.

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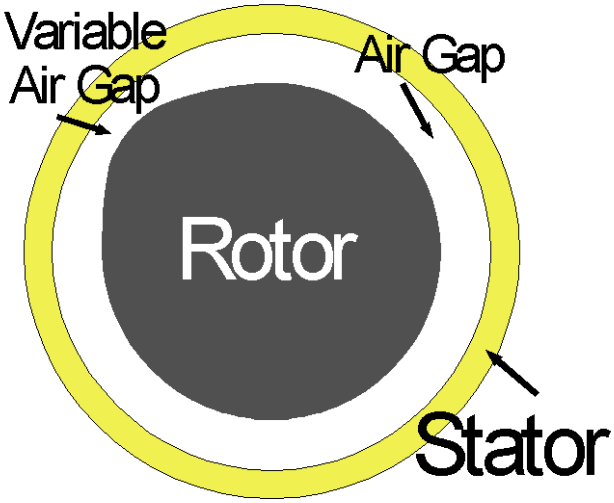
Waveform

The waveform resembles that for imbalance, because the rotor actually is out of balance. In a live, real time, display of the waveform, however, you can see the electrically induced vibration disappear immediately when you turn off the power to the motor. The signal disappearance helps you differentiate this fault from mechanical imbalance, where the vibration would coast down with shaft speed.

Rotor Bar defects can be evaluated using a current clamp and a flux coil connected to the analyzer. The motor technology program (MotorView®) calculates the severity of the rotor bar problem by looking at the spectral data from these two tools.

Dynamic Eccentricity



Spectrum

Dynamic eccentricity is similar to a broken rotor bar problem in that the variable air gap is caused by the rotor being mechanically out of round. The principal vibration appears in the spectrum at 1 x TS with sidebands spaced pole passing frequency and/or 2 x LF. Stator slot frequency may also be present with sidebands spaced at 2 x LF.

Waveform

The waveform resembles that for imbalance, because the rotor is actually out of balance. In a live, real time display of the waveform, the electrically induced vibration disappears immediately when the motor power is turned off. The signal disappearance (when power is shut off) helps to distinguish this fault from mechanical imbalance.

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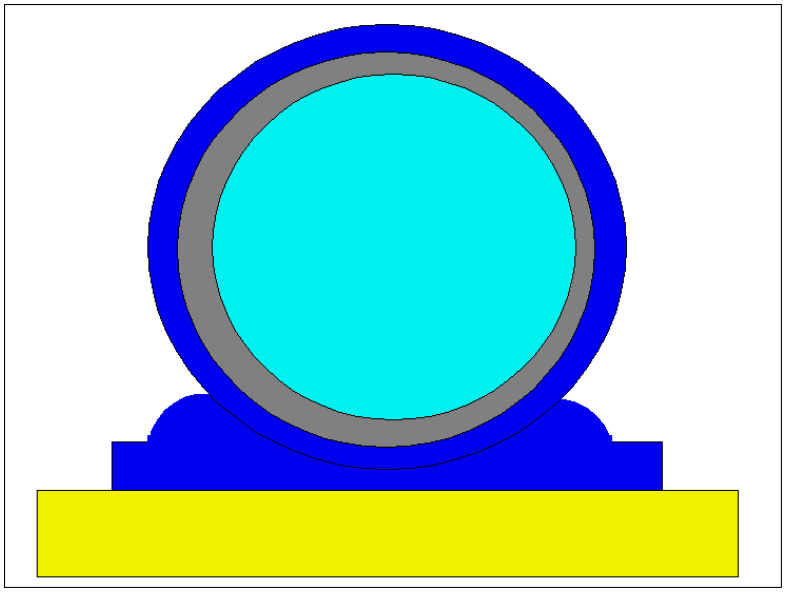
Stator Related Defects

Static Eccentricity



Spectrum

An eccentric stator will cause a fixed, unequal air gap between the rotor and the stator. This results in the spectrum having a primary peak at 2 x LF that is very directional. Look at high frequencies for slot frequency with sidebands spaced at 2 x LF. A poorly built stator, improper lamination, loose laminations, warped motor base and/or soft foot are all possible causes of this defect.

Waveform

The waveform is mostly sinusoidal in shape, with a dominant cycle at a period related to 2 x LF. As with other electrical faults, this defect can be confirmed by monitoring the signal in a live, real time waveform and securing the power. The energy should disappear immediately.

Very often, eccentric stator is caused by a soft foot condition on the motor. If soft foot is suspected, monitor 2 x LF in a live spectrum and loosen, then re-secure, one motor foot bolt at a time. Observe any change in the amplitude at 2 x LF. If the amplitude at 2 x LF is reduced during this procedure, then a motor soft foot is probable. This test does not determine which of the motor feet has the potential problem.

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Loose Stator Laminations or Loose Stator Core

Spectrum

The primary vibration appears in the spectrum at 2 x LF and harmonics of 2 x LF.

Waveform

The waveform demonstrates cycles at frequencies related to 2 x LF. This can be confirmed by monitoring the signal in a live or real time waveform or spectrum and turning off the power. The energy should disappear immediately.

Open or Shorted Windings

Spectrum

The primary vibration appears in the spectrum at 2 x LF and increases in amplitude as motor temperature climbs.

Waveform

A dominant cycle with a period related to 2 x LF is shown in the waveform. This can be confirmed by monitoring the signal in a live or real time waveform or spectrum and turning off the power. The energy should disappear immediately.

Insulation Breakdown

Spectrum

The primary vibration appears in the spectrum at 2 x LF.

Waveform

A dominant cycle with a period related to 2 x LF is shown in the waveform.

Imbalanced Phase

Spectrum

Imbalanced phase appears in the spectrum at 2 x LF.

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Waveform

A dominant cycle with a period related to 2 x LF is shown in the waveform. This can be confirmed by monitoring the signal in a live or real time waveform, or spectrum, and turning off the power. The energy should disappear immediately. Monitor the current of all three phases with a current clamp to test for this problem. The amplitude of the phase currents should be within five percent.

Loose Connector

Spectrum

The primary frequency appears in the spectrum at 2 x LF with sidebands spaced at one-third line frequency.

Waveform

A dominant cycle with a period related to 2 x LF is shown in the waveform. This problem can be particularly difficult to identify if the loose connector is intermittently making contact.

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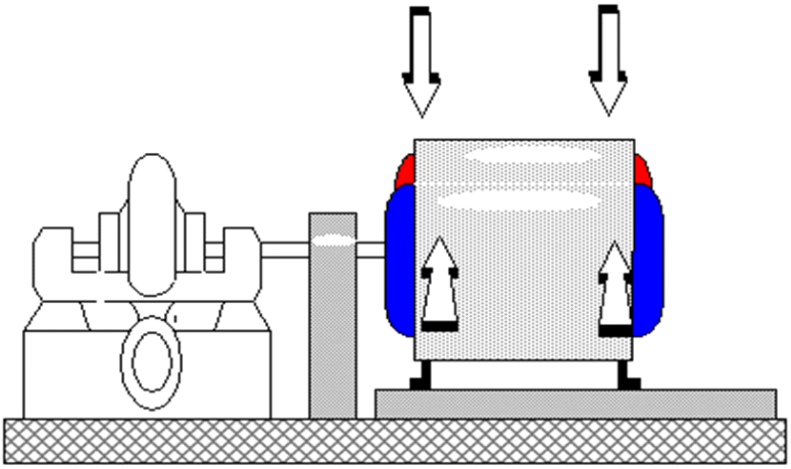
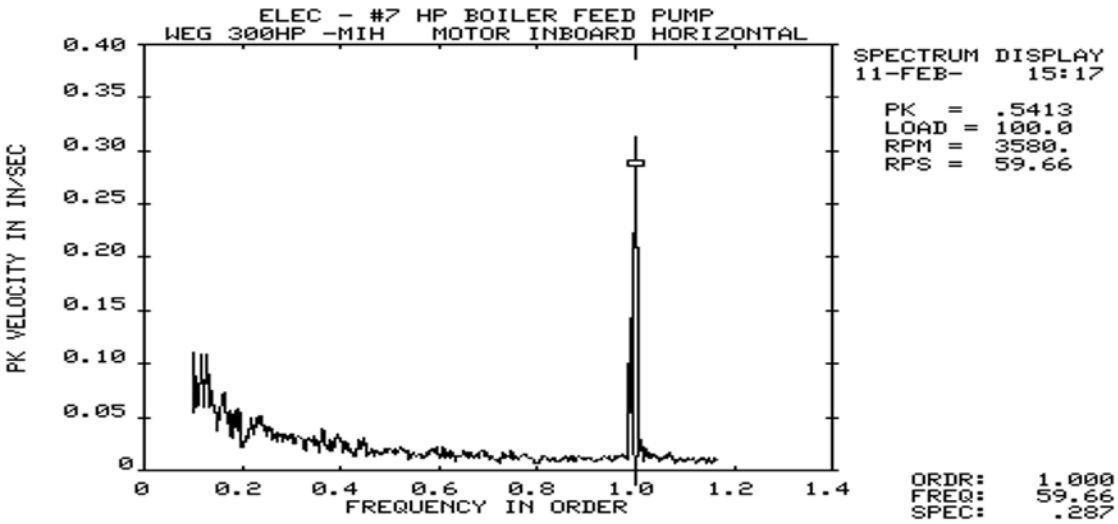
Case History – Boiler Feed Pump

300 HP, 2-Pole, 3580 RPM AC Induction Motor

- The motor supply breaker was intermittently tripping.
- Speed fluctuations of the motor were observed.
- Amperage surges of 47 amps were measured with ammeter.



The spectrum below comes from a boiler feed pump that was suspected of having an electrical fault. The vibration data was collected after the stator was tested and determined to be in satisfactory condition. The cursor is marking the running speed peak of the motor. Notice the skirt width of the running speed peak.



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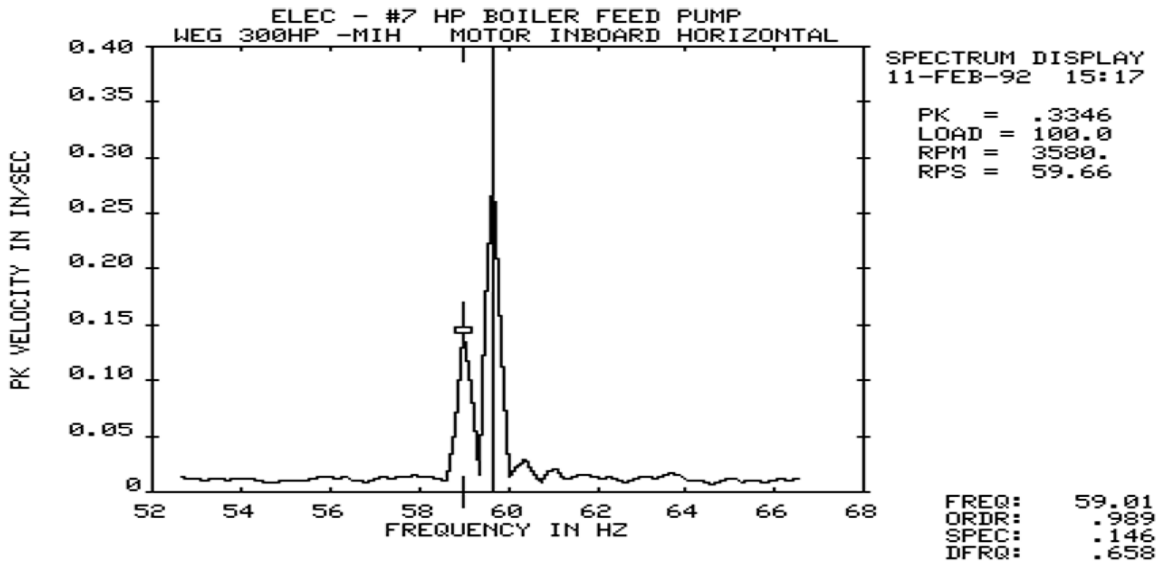
When the previous spectrum is expanded further, the sideband around motor turning speed is now more obvious. This may have been identified as imbalance if a high resolution spectrum was not taken. This demonstrates the importance of collecting high resolution data, when an electrical defect is suspected.



A mark has been set on the motor 1 x TS peak. The difference frequency of the sideband is 0.66 Hz, which equals the number of motor poles (two) times the slip frequency (0.34 Hz).

Slip frequency = .34 Hz Also, the # poles = 2

Pole Passing Frequency (# poles x slip) = .68 Hz

The sideband spacing of 0.66 Hz around 1 x TS of the motor approximates this frequency. The presence of this sideband is a good indication of a rotor bar defect. A current measurement analysis would confirm if a rotor bar problem exists. In this case, **the repair shop found five open rotor bars on this motor**.

Other useful technologies on a problem like this include the Current Analysis and Infrared technologies.

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Conclusion

This was a severe rotor bar defect because the lower slip frequency sideband is at an amplitude of about 0.15 in/sec while the turning speed is only 0.3 in/sec. Generally speaking, the lower slip sideband will need to be viewed in **db** units to see it on the display.

A rule of thumb to apply when using db units on the spectrum is: **When the lower slip frequency sideband is less than 40 db down from the turning speed peak, there is a strong probability of broken rotor bars on the motor.**

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Review Questions (Assume 60Hz line frequency for all questions)

1. What is the synchronous speed of a three phase, two pole, induction motor?

2. If an AC induction motor is running 1780 RPM, what is its slip frequency?

3. If an AC induction motor is running 1190 RPM, what is its pole pass frequency?

4. Broken rotor bars cause increase vibration amplitude at which frequency?

5. Stator eccentricity causes increased vibration amplitude at which frequency?

6. An AC induction motor is running 295 RPM. How many poles does it have?

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Gearboxes

Section 13

Objectives

- Describe basic purpose of gears and their components.
- Identify gear defects and their effect on vibration data.
- Select the best sensor, location and measurement point setup.

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General Description

Gears are commonly used in industry to provide the speed and power transmission requirements needed in industrial applications. They can also be used to change the direction of power transmission as well as provide speed, load and power transmission without slip.

Gear tooth designs have specific characteristics that can affect the measured vibration. With an understanding of a few gear design basics and terminology, troubleshooting gearboxes can be accomplished more easily.

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Gear Types

Spur Gears

Spur gears are most commonly thought of when discussing gears. The teeth are cut parallel to the shaft, and provide the highest power transmission efficiency of any type of gear design.

Spur gears are efficient with power transmission and speed changes but are noisier than other gear types.

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Helical Gears

Helical gears have teeth cut at an angle to the shaft. These gears are much quieter than spur gears. Due to the angular nature of the gear meshing and axial thrust, axial vibration is higher on these gears than on spur gears.

To avoid the higher axial thrust, double helical gears may be used. These gears, often called herringbone gears, are divided in the middle with each side having an equal magnitude angle but in an opposite direction.

If a gap exists between the two halves of the gear, then it is a double helical gear. If no gap exists and each tooth is continuous, then the gear is called a herringbone gear. The main advantage of a helical gear set is its quiet operation. The disadvantages include lower power transmission capability compared to spur gears and axial loading of the bearings.

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Bevel Gears

Bevel gears are used in right angle gearboxes to transmit power to an output shaft perpendicular to the drive shaft.

These gears use a bevel gear design to transmit the power at a 90 degree angle, so a driver can be mounted horizontally with a vertical shaft on the output of the gearbox.

Some right angle gearboxes are built with spiral bevel gears, which are quieter than bevel gears with straight cut teeth.

The advantage of a bevel drive is the ability to change the direction of power transmission. The disadvantages are inefficiency and heat generation.

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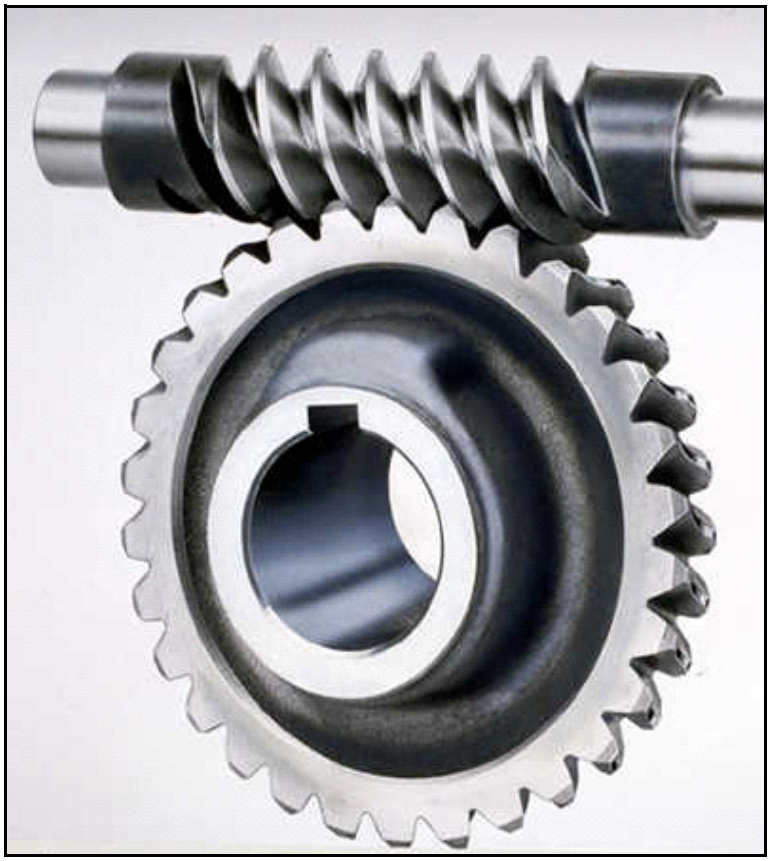
Worm Gears

Worm gears are also used to transmit power between perpendicular shafts. However, instead of using two bevel gears, a worm gear is used in conjunction with either a spur gear or a helical gear. The worm has one or more teeth wrapped around its shaft like a coil. This tooth rotates transferring the direction of power with high reduction ratios to the worm wheel, which rotates the output shaft. The advantage of a worm drive is a very high reduction ratio. The disadvantages are poor efficiency, heat generation, and low power transmission capability.



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Basic Gear Theory

The purpose of the gearbox is to provide power and or speed changes with a minimum of excessive noise and vibration. To accomplish this, the power from the drive gear must be transmitted through a line perpendicular to the common tangent and intersecting the center to center line of the drive and driven gear set.

The common tangent is a line parallel to the surface at the contact point to both of the meshing gears. This point of intersection is called the **pitch point**. The pitch point of each gear tooth must be on the center to center line between the gears in order to minimize the gear transmission error as each tooth engages its mate.

Prime Number Theory

Prime numbers are numbers that can only be divided by one or the number itself. Examples of prime numbers include 1, 2, 3, 5, 7, 11, etc. The number of teeth on each gear can be factored down to a series of prime numbers. For example, a gear with 10 teeth can be broken down to its prime factors 1 x 2 x 5, and a gear with 26 teeth can be factored into 1 x 2 x 13.

Prime numbers are important because some of the gear defects and their frequency components are calculated from an understanding of prime numbers as applied to the gear set under evaluation. When the largest prime factor common to both gears is one, one tooth on a drive gear will mesh with every other tooth on a driven gear before it re-meshes with the first tooth on the driven gear. This is called the **largest common factor** or **LCF**. If the LCF is greater than one, some teeth will mesh with others more often, leading to reduced gear life on the gear set.

Also note that machining defects and wear patterns develop and show up as defect frequencies based on the largest common prime factors between the meshing gears.

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Gear Vibration Analysis

Analysis of gear vibrations can provide a wealth of information about the mechanical health of the gears. Most defect frequencies are a result of transmission error between two meshing gears caused by machining errors, tooth deflections, looseness, eccentricity or anything that causes the power to be transferred through any point other than the pitch point.

Gear Mesh Frequency (GMF)

Gear Mesh Frequency (GMF) is the most basic gear frequency. It is calculated by multiplying the number of teeth on the gear by the turning speed of that gear. **GMF is not a defect frequency**. GMF is present in the spectrum regardless of gear condition. Its amplitude may vary depending on the gear condition and load.

GMF = #teeth x RPM

This relationship can be used to the analyst's advantage when determining the output speed of a driven gear, if the input RPM and the number of teeth on each gear is known. This is possible because any two meshing gears must have the same gear mesh frequency. **If the GMF = #teeth x RPM, then the RPM = GMF ÷ #teeth**.

When faced with the need to calculate an output speed for a single reduction gear drive, calculate the gear mesh frequency by using the input RPM and number of teeth, and divide the GMF by the number of teeth on the output gear. The answer is the RPM of the output shaft.

The RPM of the output shaft can also be calculated by first calculating the input to output ratio (#teeth in ÷ #teeth out), and then multiplying the input RPM by the ratio.

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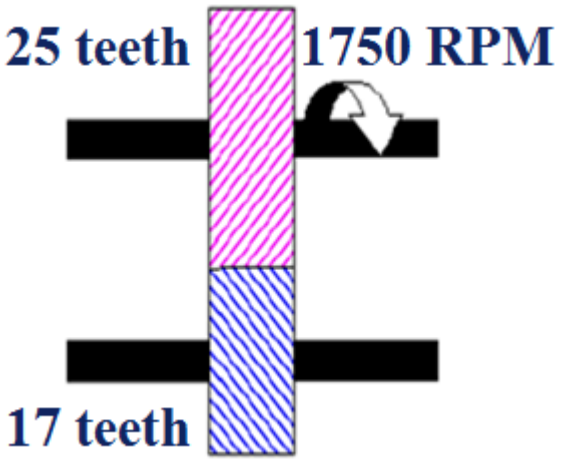
In the example below:

GMF = 25 x 1750

Output RPM = GMF ÷ 17 or

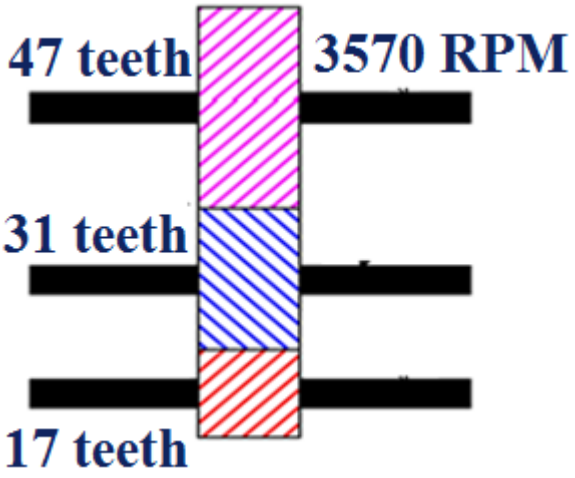
Output RPM = (25 ÷ 17) x 1750



GMF = _____

Output RPM = _____

Gearboxes with more than Two Shafts



GMF = _____

Interm RPM = _____

Output RPM = _____

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Example 1: All Gears in Mesh

Note

When all of the gears are in the same mesh, the gear mesh frequency is the same for all shafts.

In the example above, the input RPM is 3570, and the input gear has 47 teeth. It meshes with an intermediate gear that has 31 teeth. The intermediate shaft RPM is unknown.

The intermediate gear meshes with the output gear that has 17 teeth. The output shaft RPM is also unknown.

First, determine the gear mesh frequency.

$$\mathbf{GMF = \#Teeth_{in} \times RPM_{in}}$$

so

$$\mathbf{47 \times 3570 = 167{,}790\ CPM\ (or\ 2{,}796.5\ Hz)}$$

Since the gear mesh frequency is the same for all of the meshing gears, the intermediate GMF is also 167,790 CPM.

The **intermediate shaft RPM** is calculated like this:

$$\mathbf{RPM_{int} = GMF \div \#Teeth_{int}}$$

or

$$\mathbf{167{,}790 \div 31 = 5{,}412.58\ RPM}$$

Again, for the output gear, the gear mesh frequency is the same between two meshing gears.

The **output shaft RPM** is calculated like this:

$$\mathbf{RPM_{out} = GMF \div \#Teeth_{out}}$$

or

$$\mathbf{167{,}790 \div 17 = 9{,}870\ RPM}$$

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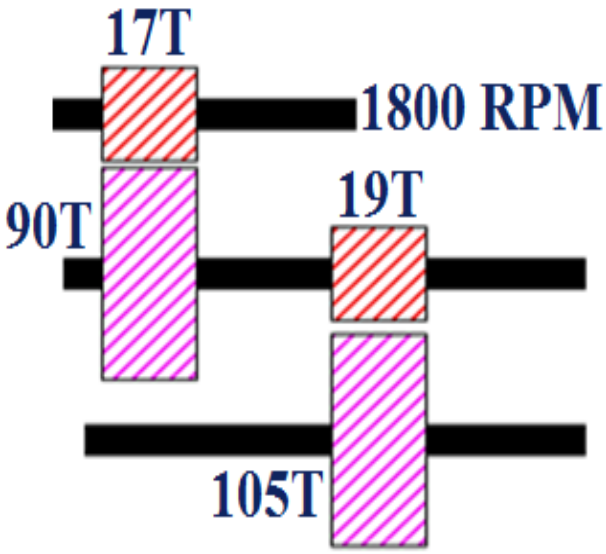

GMF1 = _____

GMF2= _____

Interm RPM = _____

Output RPM = _____

Example 2: Separate Gear Sets

In this example, which is much more common when there is more than one reduction or increase, there are two gear mesh frequencies. The input RPM is 1800, and the input gear has 17 teeth. It meshes with an intermediate gear that has 90 teeth. The intermediate shaft RPM is unknown.

The intermediate shaft has a second gear that has 19 teeth, which meshes with the output gear that has 105 teeth. The output shaft RPM is also unknown.

First, determine the first gear mesh frequency.

$$\mathbf{GMF1 = \#Teeth_{in} \times RPM_{in}}$$

so

$$\mathbf{17 \times 1800 = 30{,}600\ CPM \quad (or\ 510\ Hz)}$$

Next, calculate the **intermediate shaft RPM**.

$$\mathbf{RPM_{int} = GMF1 \div \#Teeth_{int1}}$$

or

$$\mathbf{30{,}600 \div 90 = 340\ RPM}$$

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Then, using the intermediate RPM and the number of teeth on the second gear on the intermediate shaft, calculate the **second gear mesh frequency**.

$$\mathbf{GMF2 = \#Teeth_{int2} \times RPM_{int}}$$

so

19 x 340 = 6,460 CPM (or 107.67 Hz)

Now, using the second gear mesh frequency and the number of teeth on the output gear, **calculate the output RPM**.

$$\mathbf{RPM_{out} = GMF2 \div \#Teeth_{out}}$$

or

6,460 ÷ 105 = 61.52 RPM

Worm Drives

A worm gear set consists of a shaft with a gear tooth wrapped around the shaft like a screw. This **worm gear** engages a spur gear that is called the **worm wheel**. In the case of a worm gear, the gear mesh frequency is calculated by using the output RPM and dividing by the number of teeth on the worm wheel.

Take for example a worm drive with an input speed of 1770 RPM, an output speed of 120 RPM and a worm wheel that has 48 teeth.

$$\mathbf{GMF = \#Teeth_{out} \times TS_{out}}$$

or

48 x 120 = 5,760 CPM or 96 Hz

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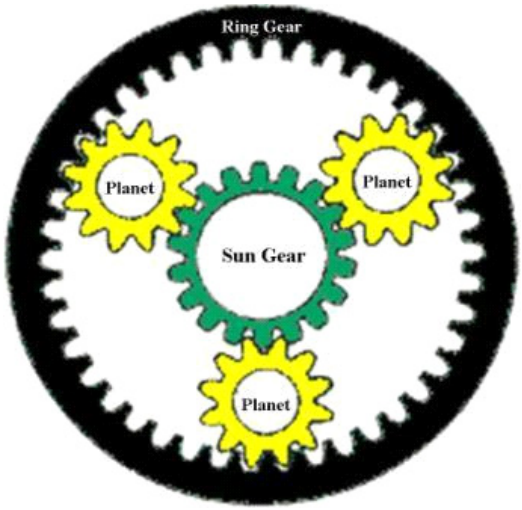
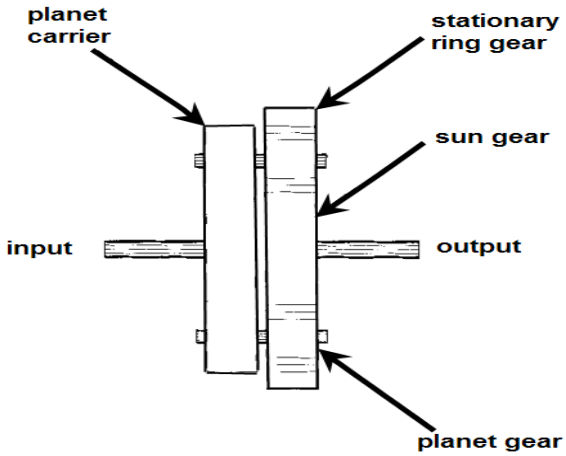
Planetary Gears





Probably the most confusing gear mesh frequency to calculate is for a planetary gear set. The input shaft is attached to the planet carrier, which drives the planet gears. The planet gears mesh with the stationary ring gear and the sun gear, which drives the output shaft.

For this gear set, the GMF is equal to the number of teeth on the planet gear multiplied by the turning speed of the planet gear. GMF is also equal to the number of teeth on the sun gear multiplied by the output shaft turning speed. Since all of the gears are in the same mesh, there is only one gear mesh frequency.

The RPM of the planet is determined by multiplying the input shaft RPM by the number of teeth on the ring gear and then dividing by the number of teeth on the planet gear.

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Gearbox Defects

Fractional Gear Mesh

Depending on the wear patterns and the Largest Common Factor (LCF) on the gear set, fractional gear mesh harmonics may appear. For example, if the LCF is two, the half gear mesh frequency peak will appear in the spectrum as the gears become worn. Likewise, if the LCF is three, then 1/3 and 2/3 gear mesh frequency peaks will appear in the spectrum.

Eccentricity also causes fractional harmonics of GMF, depending on the actual shape of the gear. If a gear has five spokes, then it is possible to have five high spots around the gear, so 1/5, 2/5, 3/5 and 4/5 GMF will appear in addition to the GMF itself.

Multiple Harmonics of Gear Mesh

Gear misalignment appears as harmonics of the GMF. Typically, the second harmonic is the most significant when trending this defect. As a rule of thumb, if the second gear mesh harmonic approaches half the amplitude of the fundamental, the gear set is likely misaligned. If the second gear mesh harmonic is higher in amplitude than the GMF itself, then it is very possible too much backlash exists in the gear set, and the gear teeth may be impacting twice during the meshing process. Check the load to evaluate this.

Multiple harmonics of GMF are typically an indication of looseness in the gearbox.

Effects of Load on GMF

The effect of load on the gear set may have two contrasting effects on the GMF amplitude. As load increases, the gear mesh frequency amplitude tends to also increase. If the load is decreased, the gear mesh frequency amplitude decreases until the light load condition causes the gears to **backlash** (bounce).

Too much backlash can also occur as the gears become worn and the clearances between the meshing gears increase. For this reason, it is recommended that whenever possible, **trend the load condition** along with the normal vibration parameters.

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Sidebands

In gear analysis, sidebands can be valuable in diagnosing gear defects. Sidebands are spectral lines that show up as equally spaced frequencies on either side of a center frequency. In this case, the center frequency would be the GMF. Sidebands appear most commonly because of wear, looseness and eccentricity of the gears.

The sideband frequency spacing is equal to the turning speed of either the input shaft or the output shaft. This spacing of the sidebands with the highest amplitudes will match the turning speed of the shaft with the faulty gear.

The severity of the problem is related to the sideband amplitudes relative to the amplitude of the peak at GMF. **When the amplitude of the sidebands is 50 percent of or greater than the amplitude of the GMF, then the problem is severe enough to warrant attention.** Sideband amplitudes equal to or greater than the GMF itself may indicate a very severe condition.

Gear Resonance

In gearboxes, eccentricity problems and excessive looseness cause the teeth to mesh with excessive force causing high levels of impaction in the machine. This causes the gear natural frequencies to be excited.

Hunting Tooth or Tooth Repetition Frequency

Hunting tooth frequency is a sub-synchronous frequency that relates to how often a single tooth on one gear meshes with a single tooth on another gear. It is calculated by multiplying the GMF by the LCF of the meshing gears, and then dividing by the product of the number of teeth on each gear.

$$\mathbf{HTF = (GMF \times LCF) \div (\#Teeth_{in} \times \#Teeth_{out})}$$

When this frequency is present, it will be very low in amplitude and may even be present as a sideband frequency. PeakVue™ does an excellent job of detecting hunting tooth problems when they are present.

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Broken Tooth

The effect of a broken tooth is very difficult to detect when using only the spectral data. If one tooth is broken, an impact will occur in the waveform once per revolution of the gear with the broken tooth. Since this occurs at the same frequency as imbalance, the effect on the spectrum may be only a slight increase in the amplitude of vibration at 1 x TS, possibly accompanied by TS harmonics.

A better way to detect a broken tooth is to examine the time waveform and look for an impact occurring at a period equal to 1 x TS. The time waveform is not sinusoidal, but has an impact and ring down once per revolution. PeakVue is also a very good tool for identifying a broken tooth defect.

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Sensor Selection

Sensor selection for gearbox analysis should consider the maximum and minimum frequencies that need to be analyzed on the gearbox.

At the high end of the frequency spectrum, the analyst will want to go out to 3.25 times the highest gear mesh frequency to include the third gear mesh frequency along with any sidebands that may be present. At the low end of the frequency range will be the hunting tooth frequency. This may require two spectral setups, one for high frequency and another for high resolution.

Set up a PeakVue measurement on each bearing of the gearbox. If the gearbox has a very high frequency requirement, you may want to select a high frequency accelerometer for your data. The PeakVue data can usually be gathered with a standard 100 mV/G sensor.

Sensor Attachment

Once the proper sensor has been chosen, ensure that the proper attachment method is used. Be aware that the frequency response of the sensor is dependent upon the mounting resonances present, due to the different mounting methods.

An accelerometer attached with a small but strong magnet may be acceptable. However, in cases involving very high frequencies, requirements above 5,000 Hz may require stud mounting to get valid vibration data.

For spur gears, the radial directions provide the best vibration information due to the direction that the forces are being transmitted through the gears. For the same reason, the axial direction is best when dealing with helical gears.

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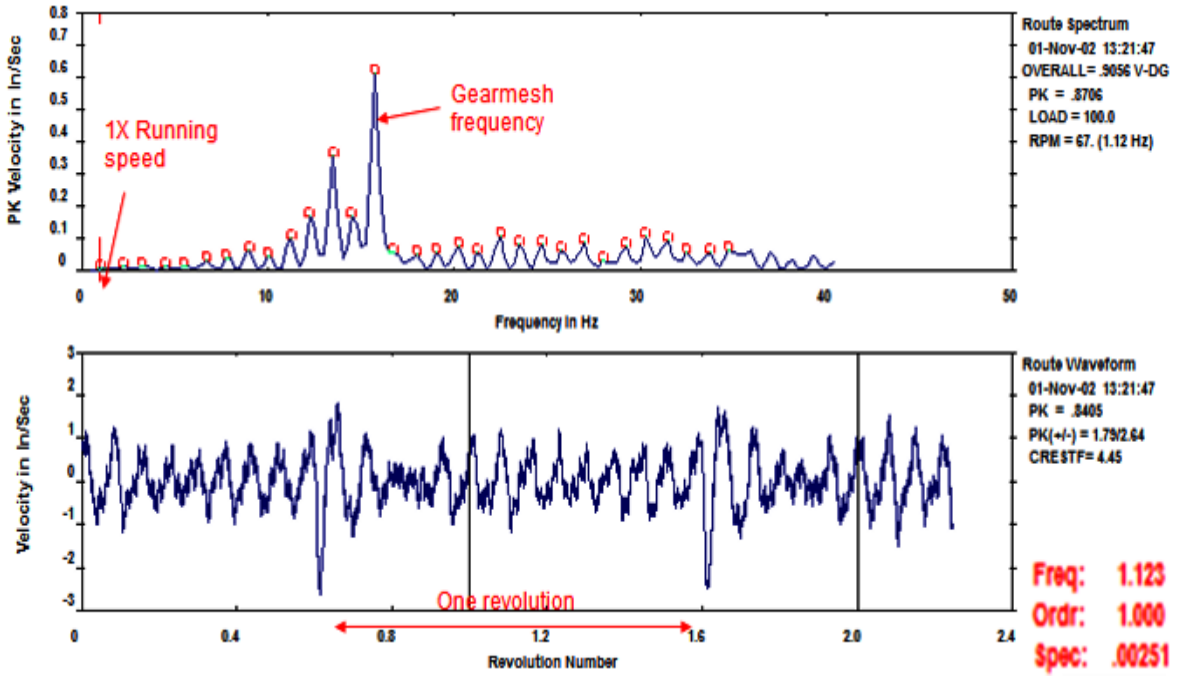
Gearbox Case History #1

100 Ton Crain Hoist Gearbox

This is data acquired using a digital recorder with the velocity spectrum indicating a very low amplitude at 1 x RPM of the input.

It looks like gear mesh frequency is the dominant force. The waveform indicates actual velocity is nearly 5 in/sec pk-pk spaced at turning speed. The low duty cycle of the 1 x low frequency energy results in a very low amplitude peak in the spectrum.



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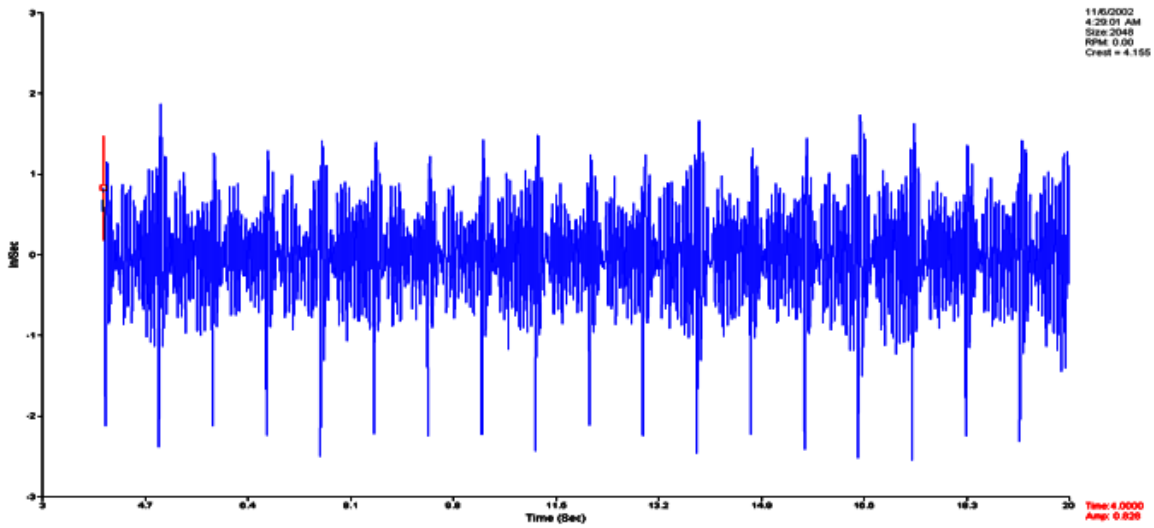
This is a longer (20sec) velocity transient time waveform from the same machine.



When the gearbox was repaired, the pinion gear driving the drum bull gear was split most of the way across at the keyway generating a high once per rev impact at once per revolution. This impact only generated a peak at .0026 in/sec (.06 mm/sec.) amplitude in the spectrum.

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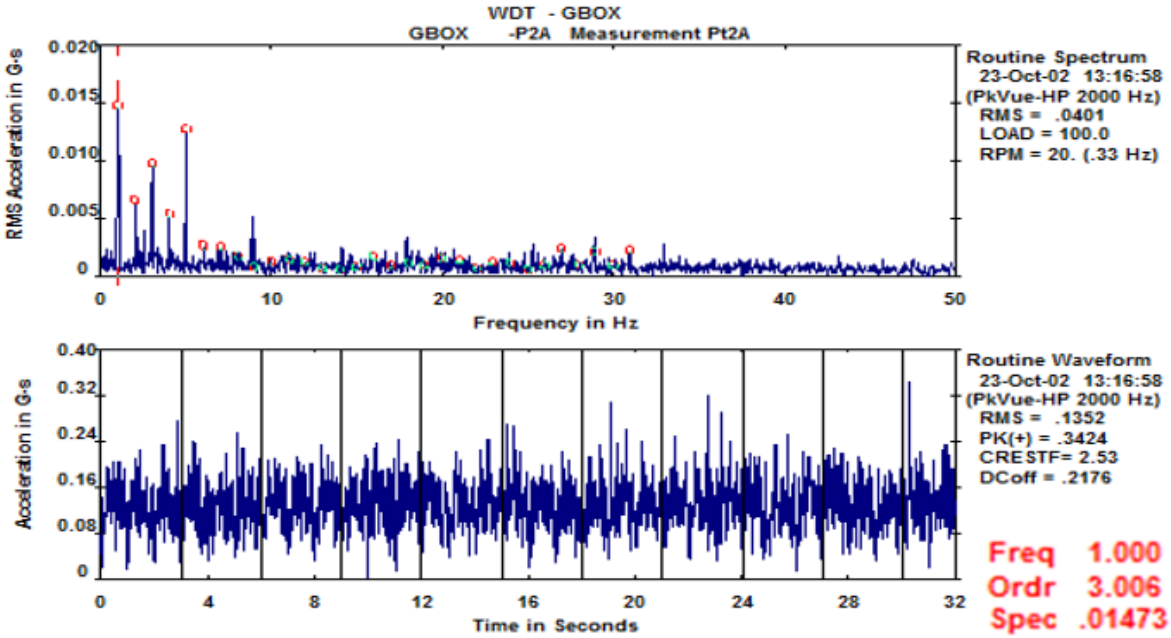
Gearbox Case History #2

Planetary Gearbox

During routine route data analysis, the analyst noticed some abnormal vibration in the spectrum. The dominant activity was at 3 x the carrier RPM. The amplitude was 0.34 g's on October 23, 2002. This was due to a defect on the ring gear. The carrier has three planet gears.



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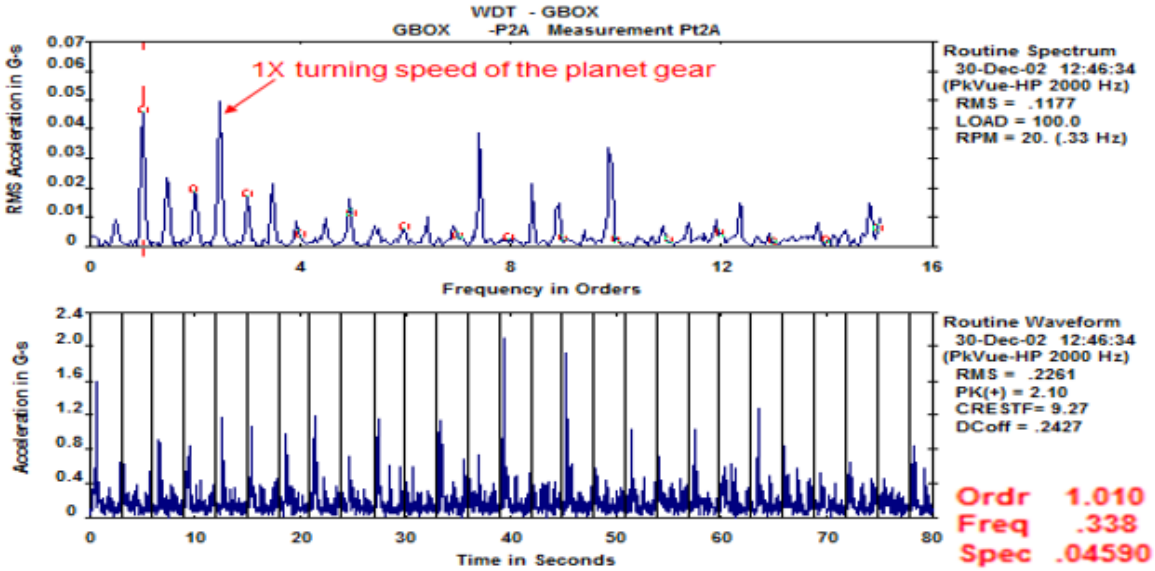
In December, the dominant peak in the spectrum is at 1 x RPM of the planet gear. The spectrum looks like it has a lot of .5 order energy (looks like looseness signature). The waveform amplitude has increased to 2.1 g's. This is six times more amplitude than in the previous data.



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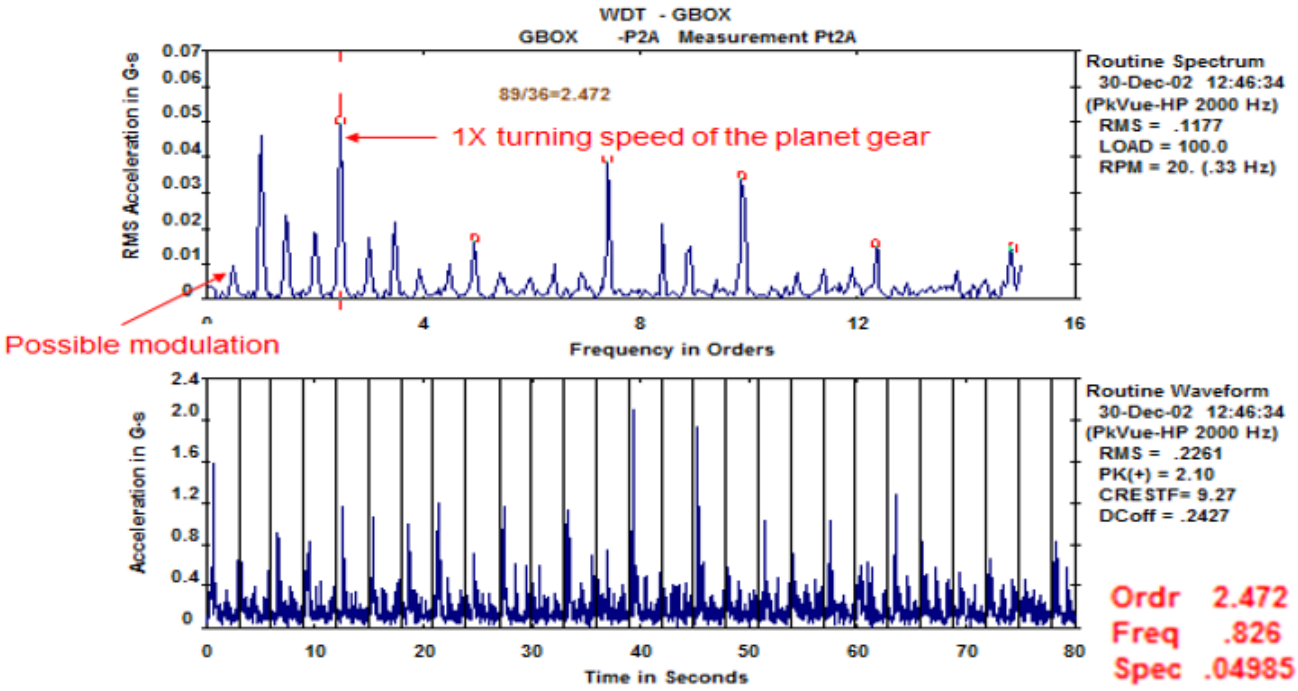
Activity highlighted is at 2.47 orders (.827 Hz.). The carrier turning speed is 20 RPM for a period of three seconds.

The planet gear turning speed is 49.58 RPM, which results in a period of 1.21 seconds. The period for two revs of the carrier is 6.0 seconds, and the period for five revs of the planet gear is 6.07 seconds. About every 6.04 seconds, both will impact at the same time generating higher amplitude creating an amplitude modulation at near 0.5 times carrier speed.



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Review Questions

1. Is Gear Mesh Frequency a defect frequency?

2. When acquiring vibration data on a gearbox, how high should the Fmax be?

3. How many Gear Mesh Frequencies are generated in a Planetary Gearbox?

4. A gearbox has an input speed of 3570 RPM. The input gear has 256 teeth and the gear that it meshes with on the intermediate shaft has 48 teeth. The second gear on the intermediate shaft has 91 teeth and the gear that it meshes with on the output shaft has 27 teeth. What type of transducer should be used and how should it be mounted?

5. What is the output RPM of the gearbox in question number four?

6. What spectral pattern is usually associated with gearbox defects?

7. What spectral pattern is often associated with gearbox misalignment?

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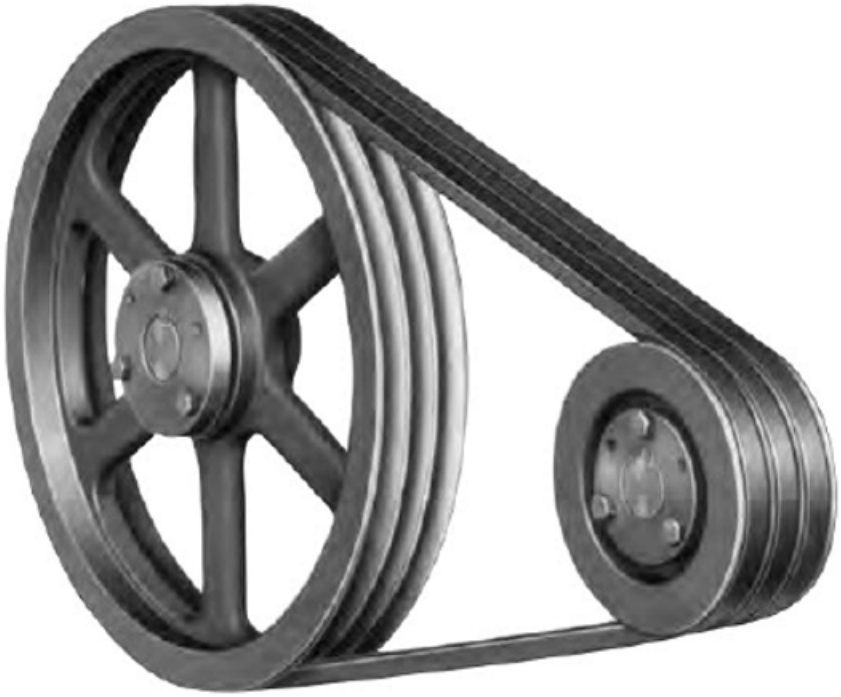
Belt Vibration

Section 14

Objectives

- Overview of common belt design and operation.
- Calculate belt frequencies.
- Describe belt related faults detectable with vibration.
- Describe the vibration characteristics of belt related faults.



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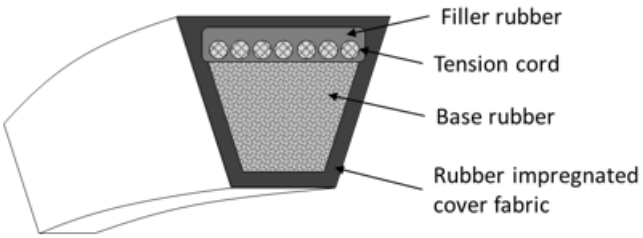
Overview of Belt Design and Operation

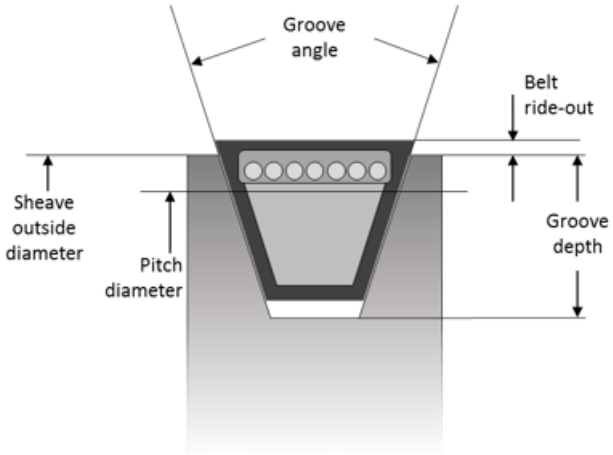
A significant portion of mechanical systems in use today are composed of belt-driven machinery.

In order to better recognize and correct belt problems, it is important to have knowledge about belt design and understand the operational characteristics.

Belts come in many shapes and sizes to accommodate most power transmission requirements. The V-belt is the most common belt applied in industry. V-belts can operate at speeds up to 6600 feet per minute. Cog belts and Ribbed V-belts can run much faster. However, all belts operate on the principle of friction.

The V-belt has straight sides. As the belt is bent around the sheave, the bottom half of the belt compresses, causing the sides of the belt to bulge and wedge tight against the grooved walls of the sheave. While the bottom half of the belt compresses, the top half stretches under tension. Only the center part of the belt does not change shape as it passes around the sheave. This center part is reinforced with one or more rows of cord strands that enable the belt to carry loads. The material is based on the load to be carried. Cord material may be rayon, nylon, glass, polymers or steel. The cords are encased in a harder rubber for strength and stability.





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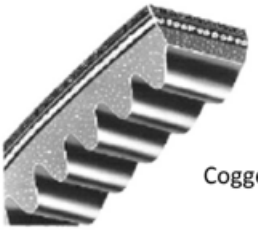
A variation of the standard V-belt is a cog belt. The cog belt allows for more flexibility in the compression section of the belt, allowing it to be used on smaller diameter sheaves. A cog belt can run at higher speeds compared to a standard V-belt of the same size.

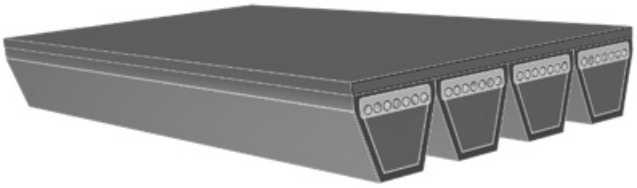


Cogged V-belt

Another variation of the V-belt is the ribbed V-belt, which tends to operate quieter and can be used to operate at speeds up to 10,000 feet per minute. It is wider and thinner than a standard V-belt and has several small V's that ride on a specially designed sheave. This is the only type of V-belt that should contact the bottom of its sheave during normal operation.

Ribbed V-belt

The Banded belt is sometimes used in place of multiple belts. The Banded belt is several V-belts sharing a common back that connects the belts along the top.



Banded V-belt

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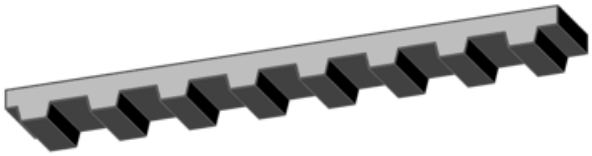
Benefits of a banded belt:

- matched sets
- no lateral movement
- no belt turnover
- uniform slippage across all segments

A double sided V-belt is essentially two single V-belts molded back to back. The double sided V-belt is used on drives with reverse bins. Their power is transmitted by both sides of the belt.

Double sided V-belt

Timing belts or positive drive belts are used when exact speed is required and no slippage can be allowed. The timing belt has special teeth, similar to gear teeth and must be used over a matching gear tooth sheave.



Timing belt

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Belt Length Standards

There are three industry standards for measuring belt length.

L-Series Belts

Fractional horsepower L-series belts use the length of the outside as the numbering system. For example, a 4L215 belt is 21.5 inches long around the outside circumference.

A, B, C, D Series Belts

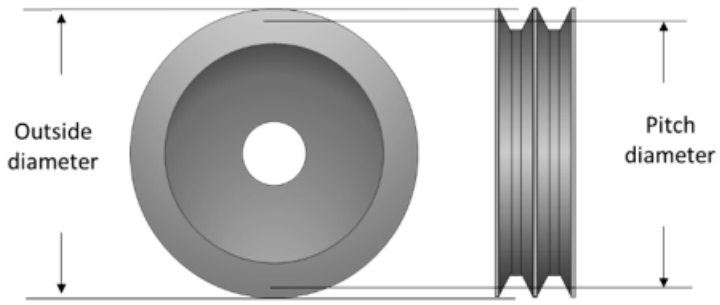
The A, B, C, D series belts use the inside length as the numbering system. For example, a B48 belt is 48 inches along the inside circumference. It would be 51” along the outside circumference.

3X, 4X, 5X Series Belts

The 3X, 4X, 5X series belts use the pitch length measurement for the numbering system. The pitch length is the distance along the belt in the region of the reinforcing cord. This length is considered to be the nominal length of the belt. Pitch length should be used when making any belt calculations.

Sheaves

Sheaves have two standard measurements, the outside diameter and the pitch diameter. The pitch diameter would be measured across the sheave to the pitch line of a belt in the groove. Use the pitch diameter for belt frequency calculations.



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Four Variables that Affect Belt Power Transmission

Belt drives are friction drives and depend on friction between the belt and the sheave to transmit power to the driven sheave. However, friction generates heat, and heat is a hazard to belts and belt life. There is a fine line between just enough friction and too much friction.

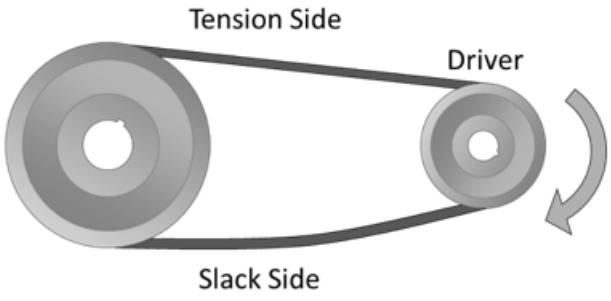
The ability to transmit power depends on friction, belt tension, belt wrap and belt speed. The speed is usually measured in feet per minute. Belt speed can also be calculated and expressed in CPM, Hz or Orders of turning speed when diagnosing belt related vibration problems.

Friction

The presence of surface oil, grease and heat will damage belts and are the most common sources of efficiency losses. The friction is affected by belt tension which must be enough to force the belt into the groove and hold it tightly against the grooved walls. The belt should not contact the bottom of the sheave groove. This will cause the sheave to push the belt out and away from the sheave walls. The proper belt tension is the least amount of tension that will keep the belts from slipping. Too much tension accelerates the wear on the belts and sheaves and puts extra load on the bearings.

Belt Tension

When the sheaves are static, the tension throughout the belt is constant. As the drive sheave rotates, it pulls on the belt, increasing belt tension to the driven sheave. This is often referred to as the tension side of the belt. The belt on the opposite side, or the following side, then tends to be somewhat looser and is called the slack side.



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Belt Slip

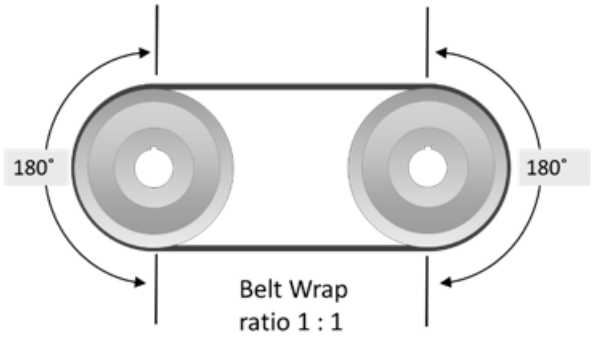
When there is excessive load on the belt, the difference between the tension and slack side can cause the belt to pull away from the sheave and allow the sheave to slip. This is referred to as **belt slip**. Belt slip generates tremendous heat causing slip burn on the belts, which substantially shortens belt life and wears the sheave.

Belt slip occurs in all belts **except** the positive drive timing belts.

Another condition that results from the high loading of belts is **belt creep**. Since the tension side of the belt is stretched, the driving sheave receives more belt than it gives out. This distance creates relative motion between the sheave and the belt, which is referred to as belt creep. It is present in most belts, except those which have internal cords that do not stretch, such as steel corded belts. Even under the best conditions, belts slip. In multiple belt drives they slip and creep at different rates generating excessive amounts of heat.

Belt Wrap

Belt wrap affects the friction, therefore the efficiency, of the belt drive. Belt wrap, or the arc of contact, is the distance around the sheave that the belt contacts the groove. The greater the belt wrap, the greater the friction. The best belt wrap is a wrap of 180 degrees, half the circumference of the sheave, which can only be achieved with a ratio of 1:1 between the sheaves where both sheaves are the same diameter. A ratio of 1:1 could be the worst ratio from a vibration standpoint, and most power transmission requirements are not a one to one ratio.



A ratio of 5:1 has much less belt wrap, and will therefore have more belt slip. A ratio of 3:1 has acceptable belt wrap when the center to center distance is great. If the center to center distance is too short, the belt wrap is substantially reduced, decreasing friction and increasing belt slip.

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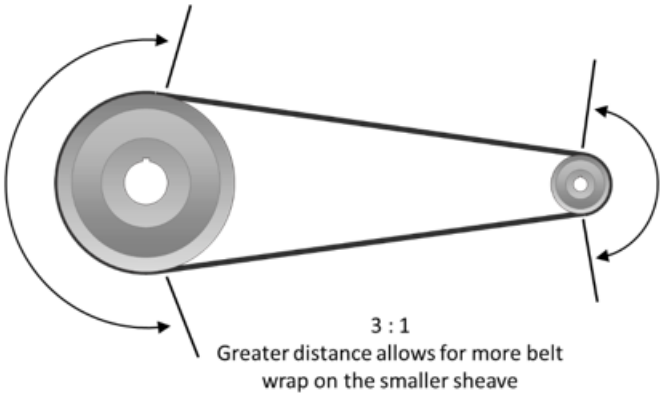
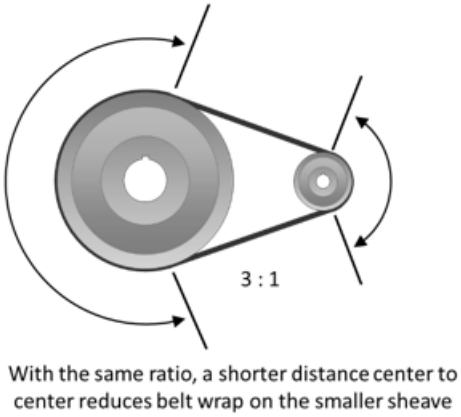
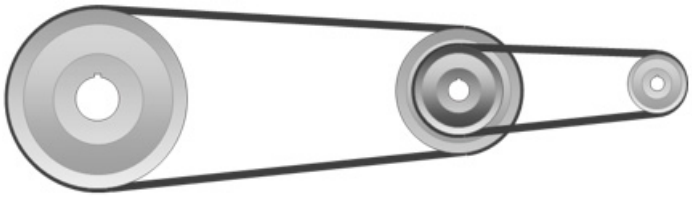
For applications where the center to center distance between the sheaves is close, a ratio greater than 3:1 is not recommended.





If a higher ratio is required, use a stepped drive with an intermediate shaft.



Stepped drive with intermediate shaft

Belt Speed

The fourth variable that affects the belts ability to transmit power is the speed of the belt. This is usually measured in feet per minute. That is, the number of feet that a position on the belt travels in one minute's time. Or, how many feet of belt pass a fixed position. Typically, the faster the belt is moving, the more efficient the system is, and less tension is required. However, there are some inconsistencies that must be considered. For example, as the belt rotates faster, centrifugal force pulls outward on the belt pulling it away from the sheaves. This increases the possibility of belt slip.

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Belt drives can be very efficient with efficiency ratings sometimes above 95 percent. But improper tension, belt wrap, improper application, poor installation, mismatched sheaves to belts, poor machining and speed can reduce the friction and severely reduce the efficiency.

Due to their design, some belts generate more vibration than others. Here, we provide a listing of the belt types rating them on a scale of one to 100, with 100 being the smoothest and quietest operating from a vibration standpoint.

Belt Types and Ratings

Endless Woven Flat belt	100
Poly V-belt	60
Link belt (without rivets)	50
Riveted Link belt	40
Wide angle V-belt	30
Timing belt	25
Banded V-belt	12
Standard V-belt	10

The lowest rated of these belts is the standard V-belt, and is the most common in industrial applications. It is most common partly because it will stay in the sheave and transmit power with minimal effort or costs to provide precision operating conditions. For example, to upgrade to a poly V-belt means replacing all the sheaves in a drive and ensuring precision sheave alignment (the poly V-belt is less tolerant of sheave misalignment than the standard V-belt). Upgrading to this system can be cost prohibitive.

Heat creates problems for belts. Most belts should not run in applications where the ambient temperature will be over 140 degrees F. The cooler the belt operates, the longer its useful life.

Belt guarding should include cage type expanded metal (or something similar), to allow plenty of airflow to cool the belts. Solid guard enclosures trap heat, causing the belts to overheat, shortening the life. Banded belts tend to prevent heat dissipation by trapping it.

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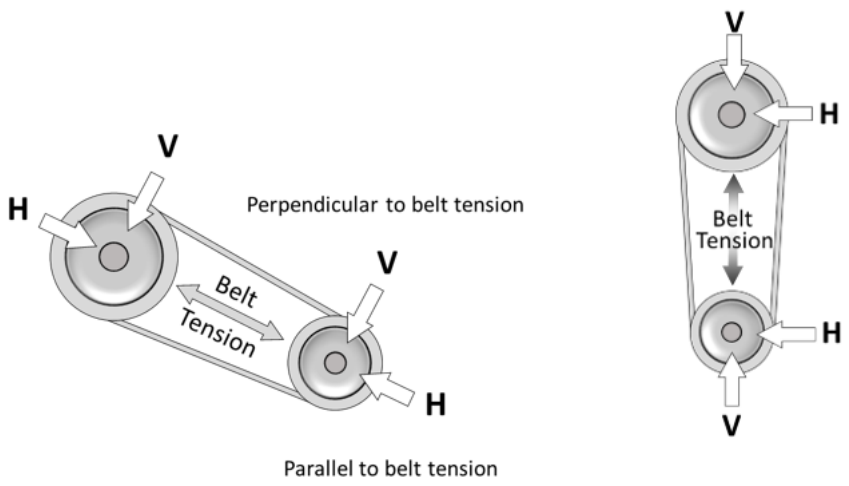
Measurement Point Orientations for Belt Drives

Belt drives can be oriented in any direction. Setting a standard measurement convention for all monitored belt drives may increase accuracy in detecting belt related faults.

Since the belt reaction forces are usually in-line with the belts, the most optimal vibration measurement orientation for detecting belt defects is in-line (parallel) with the belt tension and perpendicular with the belt. These are radial measurements (horizontal and vertical). Axial measurements will help detect additional belt related problems such as misalignment.

For non-horizontally mounted equipment, the measurement point description can be edited to include the orientation of the sensor to the belt. This could help during data analysis.



For example, when the belt tension is in the vertical direction, consider the following:

ID	Description	RPM
M1H	Motor OB Horz - Perp to Belt	1785.0
M1P	Motor Outboard Horz Peakvue	1785.0
M1V	Motor OB Vert - Inline to Belt	1785.0
M1A	Motor Outboard Axial	1785.0
M2H	Motor IB Horiz - Perp to Belt	1785.0
M2P	Motor Inboard Horz Peakvue	1785.0
M2V	Motor IV Vert - Inline to Belt	1785.0
M2A	Motor Inboard Axial	1785.0

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Belt Problems and Vibration Data

Note

Most of the belt problems discussed in this section will reference V-belt drive systems.

Belts typically have a designed life of approximately 25,000 hours, or about two years and 10 months. Sheaves commonly have a design life of 5:1. That is, they should be replaced after each fifth belt set is used. In industry today, it is common for belts to wear out much sooner, but many times it is assumed that the sheave will last indefinitely. When belts do not reach their expected life, the cause can be due to poor design, poor installation techniques or even due to worn sheaves.

Since a wide variety of problems can occur during the installation and assembly of the belt drive, diagnosing a specific problem can be difficult.

Some causes of belt related vibration are listed below.

- Belt defects (cracks, broken or missing pieces, hard (burned) spots, etc.)
- Shaft misalignment
- Sheave groove misalignment
- Belt resonance
- Belts too tight
- Belts too loose
- Sheave eccentricity
- Sheave hub cocked
- Bent shaft

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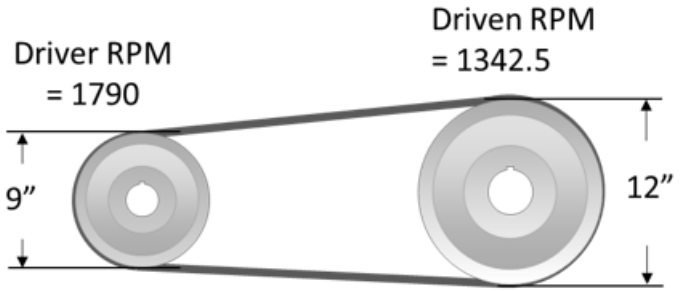
Vibration Frequencies in Belt Drives

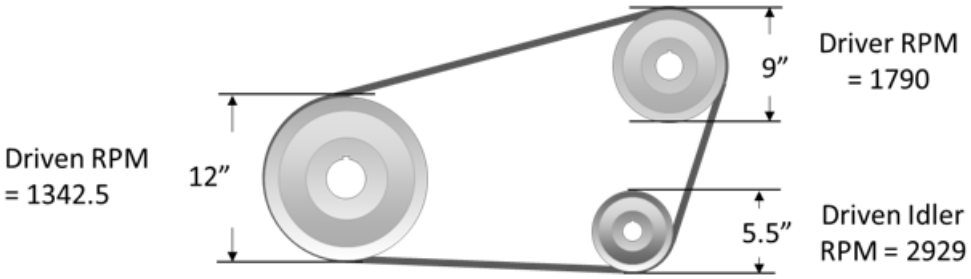
When diagnosing belt related vibration data in a two sheave system, there will be at least three primary frequencies of interest.

- Belt operating frequency
- Drive sheave operating frequency
- Driven sheave operating frequency



Common two sheave driver/driven system

For belt drive systems containing more than two sheaves, the operating speed of each sheave will be essential for analysis, along with the belt operating frequency.



Multiple sheave system
RPM of each shaft should be known for analysis

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Calculating Belt Frequencies

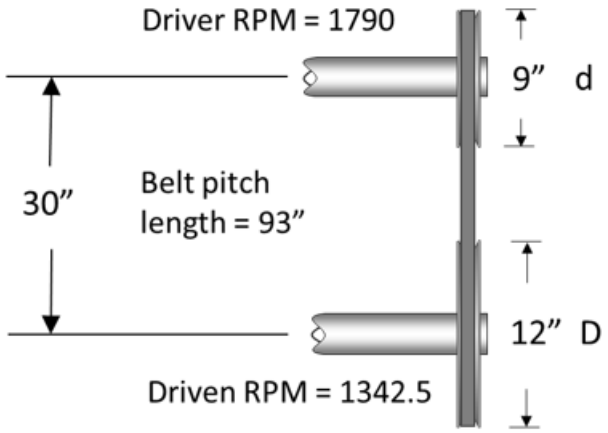
Several of the belt drive vibration problems listed previously do not cause vibration at calculated belt frequencies.

The primary reason for calculating belt frequency is to help identify frequency patterns that characterize worn or defective belts. Information such as belt length, sheave diameters and shaft speeds will be necessary to calculate the belt operating frequency using the equation below.

$$Belt\ Frequency\ in\ cpm = \frac{3.1416 \times Sheave\ Diameter \times Shaft\ RPM}{Belt\ Length}$$

Note

When using this formula, use the sheave diameter and RPM from the same shaft.



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Calculate belt frequency in CPM using the values from the previous image.

$$Belt\ Frequency\ in\ cpm = \frac{3.1416 \times 9 \times 1790}{93}$$

$$Belt\ Frequency\ in\ cpm = \frac{50611.2}{93}$$

$$Belt\ Frequency\ in\ cpm = 544.2\ cpm$$

If the belt length is not known, it may be calculated using the formula below. Use this formula to find the belt pitch length as described earlier in this section.

$$L = 2C + 1.57(D + d) + \frac{(D - d)^2}{4C}$$

Where:

L = belt pitch length (in inches)

2 = constant

C = center to center distance between sheaves (in inches)

1.57 = constant ($\frac{\pi}{2}$)

D = large sheave diameter (in inches)

d = small sheave diameter (in inches)

4 = constant

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Calculate the belt pitch length using the values in the previous image.

$$L = 2C + 1.57(D + d) + \frac{(D - d)^2}{4C}$$

$$L = (2 * 30) + 1.57\ (12 + 9) + \frac{(12 - 9)^2}{4\ *\ 30}$$

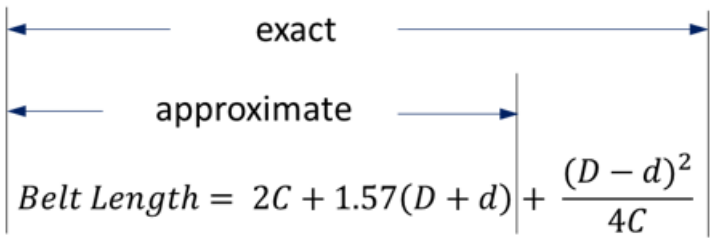
$$L = 60 + 1.57\ (21) + \frac{(3)^2}{120}$$

$$L = 60 + 32.97 + \frac{9}{120}$$

$$L = 92.97 + .075$$

$$L = 93.045$$

The belt pitch length formula can be used as an approximate, or exact calculation.



Ways to Obtain Belt Frequency

- Belt operating frequency can also be detected with a strobe light if the guarding allows line of sight to the belt.
- A photo tach can be used, and will be very accurate, but requires setup and reflective tape to trigger the tach.
- A laser tach (speed sensor) is also a good tool for measuring an accurate belt RPM.

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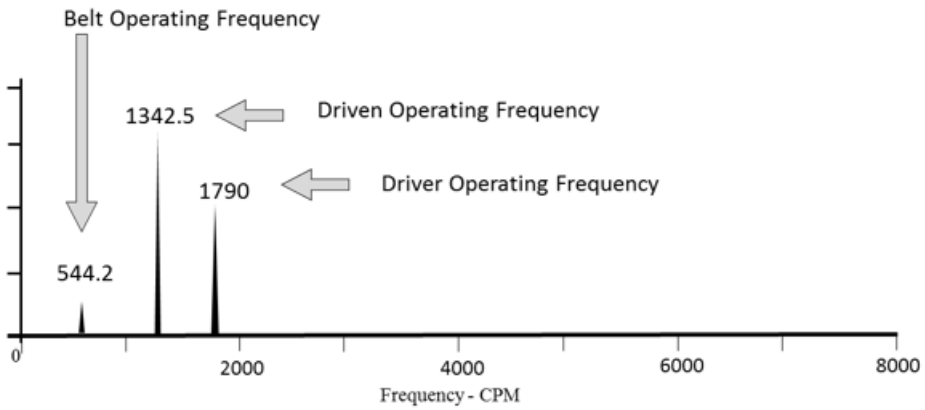
Appearance of Belt Operating Frequency in the Frequency Spectrum

Belt operating frequency will be below the RPM of the driver or driven shafts in the machine. Therefore, belt frequency is sub-synchronous to the all sheaves in the system.



Recognizing Belt Problems in the Vibration Data

Belt related vibration can be traced to two sources, belt defects and operating problems. The problem will either be defects on the belt itself, such as cracks, worn or deformed surfaces, etc, or the problem will be related to installation or operating parameters such as misalignment, tension, eccentric sheaves or a bent shaft, etc.

Belt Defects

Defects on the belt will produce vibration frequencies at belt frequency and harmonics of belt frequency, while installation and operating problems will produce vibration at the operating frequency of the sheave associated with the problem.

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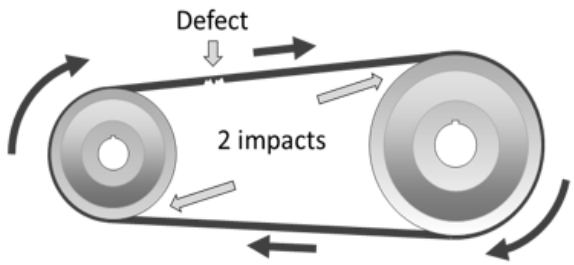
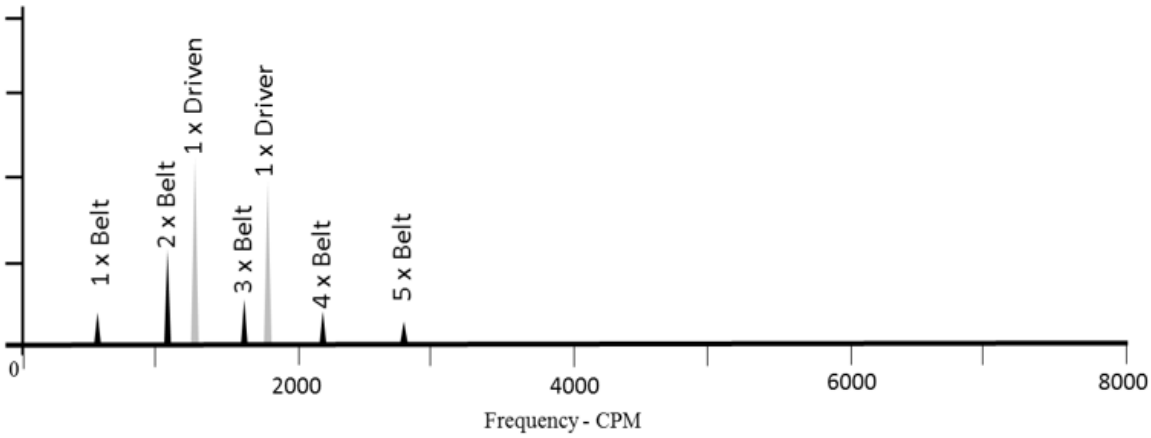
Spectral Characteristics of *Defective Belts*

A defect on a belt will impact each sheave in the system once per belt rotation. In a two sheave system, this causes an impact frequency at 2x the belt frequency. In the spectrum, the dominant frequency would often be 2x belt frequency due to the defect impacting each sheave per belt rotation. When a defect is present, there will normally be harmonics of the belt frequency in the spectral data.



The defect will impact each sheave in the system. This produces an impact frequency of 2 x the belt frequency, along with harmonics of belt frequency



2x Belt Frequency and Harmonics of Belt Frequency

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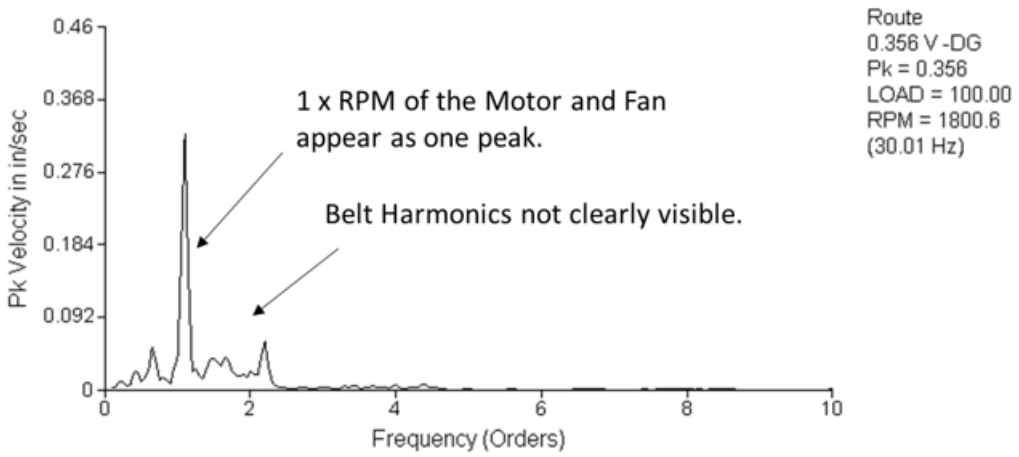
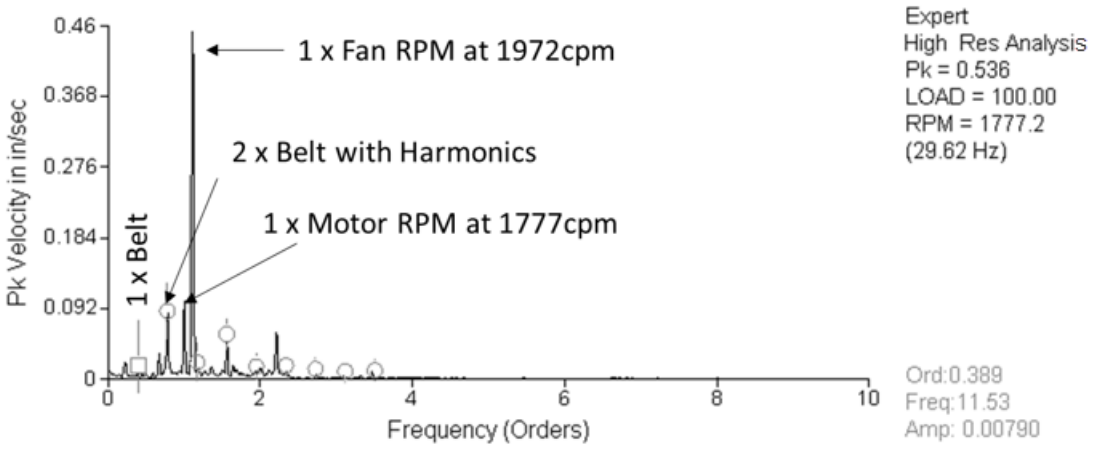
Often when a belt defect is present, the belt harmonic peaks will be difficult to see in the spectrum because the harmonic frequencies can be very close to the peaks that appear at 1 x RPM of the driver and driven units. This can make belt defect analysis somewhat difficult.



With normal settings in the AP set for route measurements, belt defects can be difficult or impossible to see in the data.

To separate the peaks in the spectrum, take a radial measurement in line with the belt using a high resolution setting and a lower Fmax setting. For example, 3200 lines of resolution with an Fmax of 500 Hz.



With more lines of resolution, the belt operating frequency and harmonics are clearly visible along with 1xRPM of both the motor and the fan.

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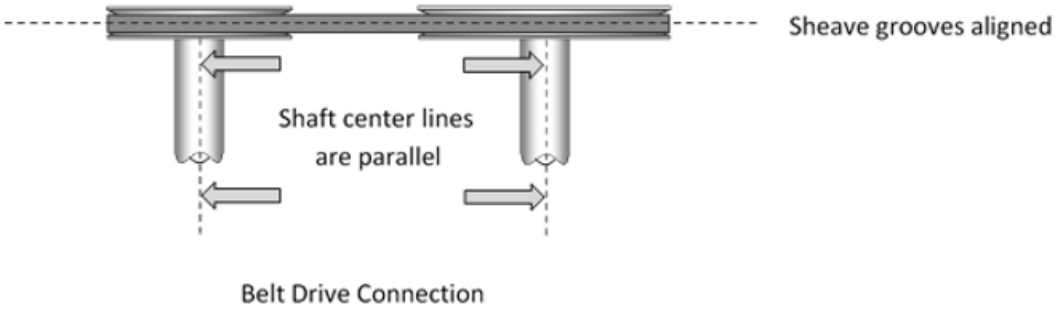
Installation and Operational Problems

Sheave Misalignment

For belt driven equipment, the shafts should be parallel and the sheave grooves should be aligned to allow for a straight belt path.



Allowing rotating equipment to operate in a misaligned condition can lead to excessive power consumption, accelerated component wear, and sometimes the potential for catastrophic equipment failures.

Misaligned sheaves cause a reduction in the efficiency of the machine.

As with coupling misalignment, there are various types of sheave misalignment. In most cases, the misaligned condition will include a combination of the different types.

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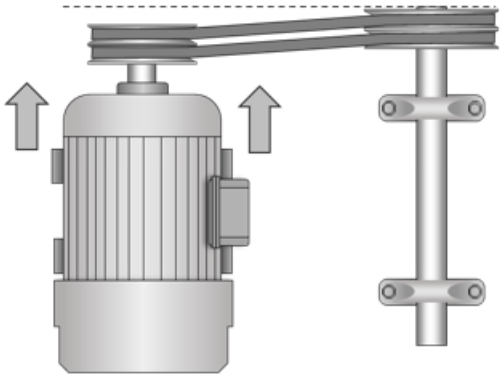
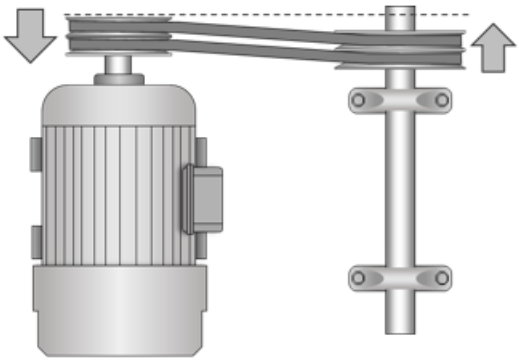
Offset (Parallel) Misalignment

Shaft centerlines are parallel, but sheaves are offset from the position of either the driver or driven component, or by sheaves mounted offset to one another on the respective shaft. Correction for parallel alignment is not as complex as other misalignment conditions.



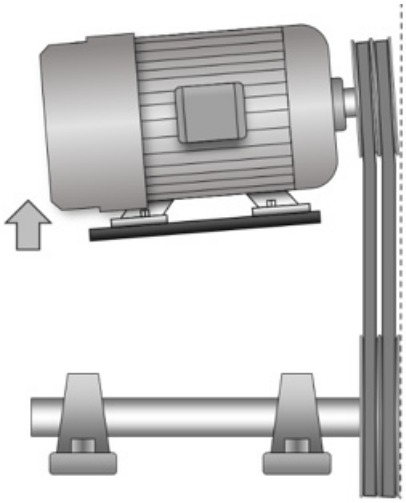
Shaft center lines are parallel, but the motor is offset to driven component



Shaft center lines are parallel, but sheaves are not positioned correctly on the shafts

Vertical Angular Misalignment

Shafts are not parallel in the vertical plane. Correction requires moving either end of the driver or driven component in a vertical direction. In most cases, the driver is adjusted for alignment correction.



Shaft center lines are misaligned vertically

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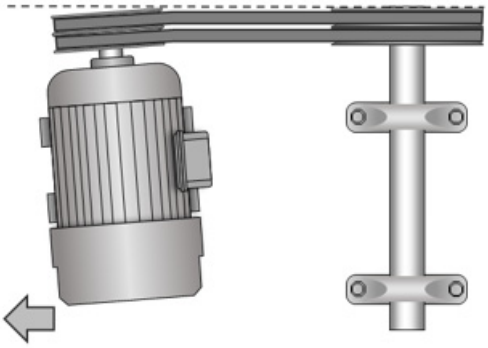
Horizontal Angular Misalignment

Shafts are not parallel in the horizontal plane. Correction requires moving either end of the driver or driven component in a horizontal direction. The driver is most commonly the adjusted component.



Shaft center lines are misaligned Horizontally

Combination of Vertical and Horizontal (Twisted)

Shafts are not parallel in either plane. Correction requires moving either end of the driver and/or driven components. The driver is most commonly the adjusted component.

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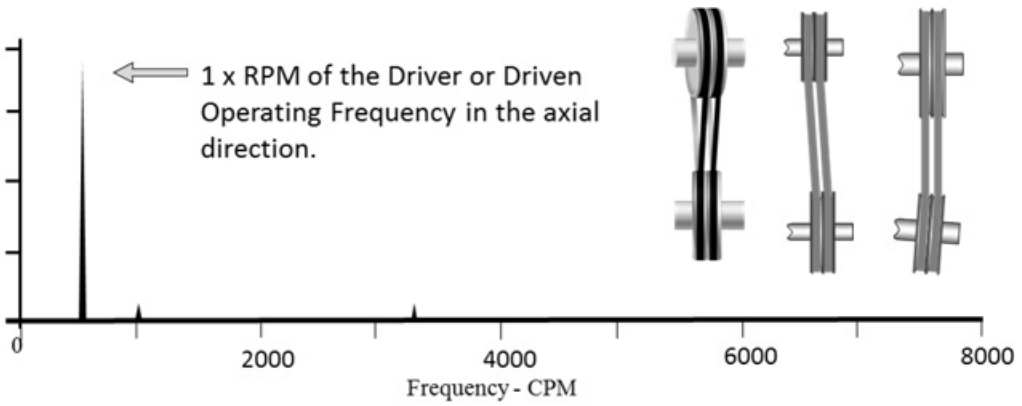
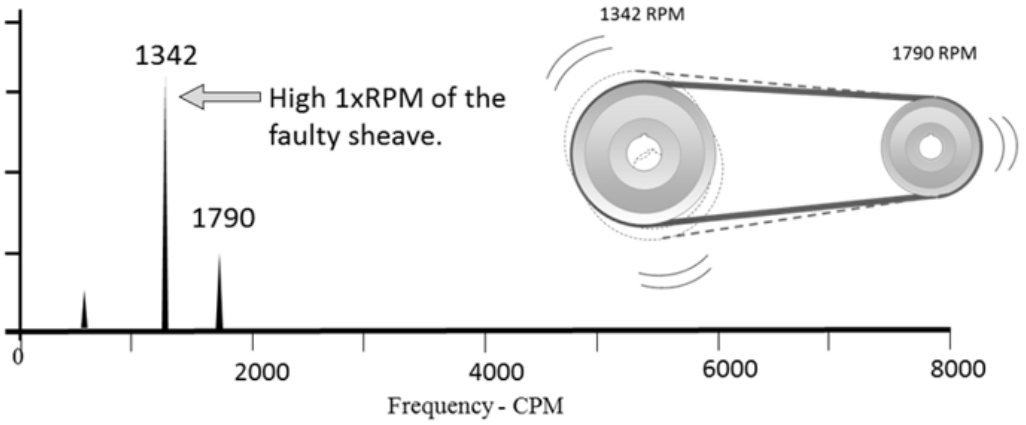
Special Characteristics of Belt/ Sheave Misalignment

In any of the misaligned conditions, the belt is forced into the sheave groove at an angle. The belts ride up on the groove wall and slap back into the groove. This force is a continuous pull on the sheave in the **axial** direction. This constant force produces a high vibration in the axial direction at the operating frequency of the sheaves in the system.



Spectral Characteristics of Eccentric or Imbalanced Sheaves

An eccentric sheave, or an imbalanced sheave can cause high vibration at 1x RPM of the faulty sheave. A bent shaft at the sheave produces the same results. The highest amplitude will be in line with the belt, which is a **radial** measurement. The high 1xRPM of the faulty sheave is usually present in the data of both the driver and driven components.



Eccentric or imbalanced sheave causing high 1 x RPM at the frequency of the faulty sheave in the radial direction

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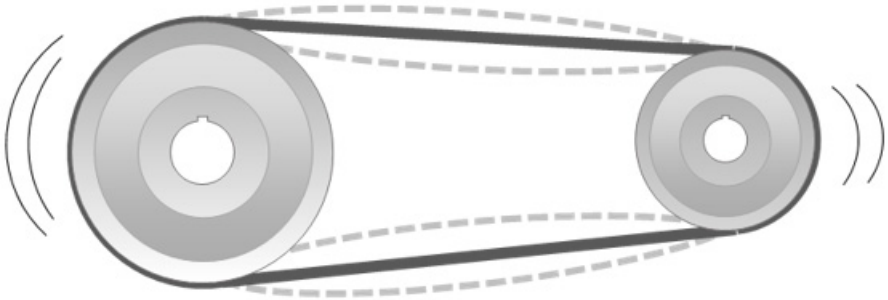
Vibration Characteristics of Belt Strand Vibration

Almost all of the faulty conditions mentioned in this section can, and usually do, cause belt strand vibration. The frequency and the effect of belt strand vibration will vary depending on properties of the belt such as tension, length, size, center to center distance between sheaves, etc. Structural properties of the machine, stiffness and flexibility of the system, will affect the belt strand vibration.



Belt strand vibration can be random and/ or periodic. Loose tension can cause the belts to vibrate randomly at a low frequency, while high tension may cause the belt to react to the fault at a higher frequency. With multiple belts in one system, each belt can have a different strand frequency depending on the tension of each belt and other characteristics mentioned above. When belt strand vibration is significant, the random nature of a belt vibrating between the sheaves often produces a pattern of amplitude and frequency variations through out the measurement.

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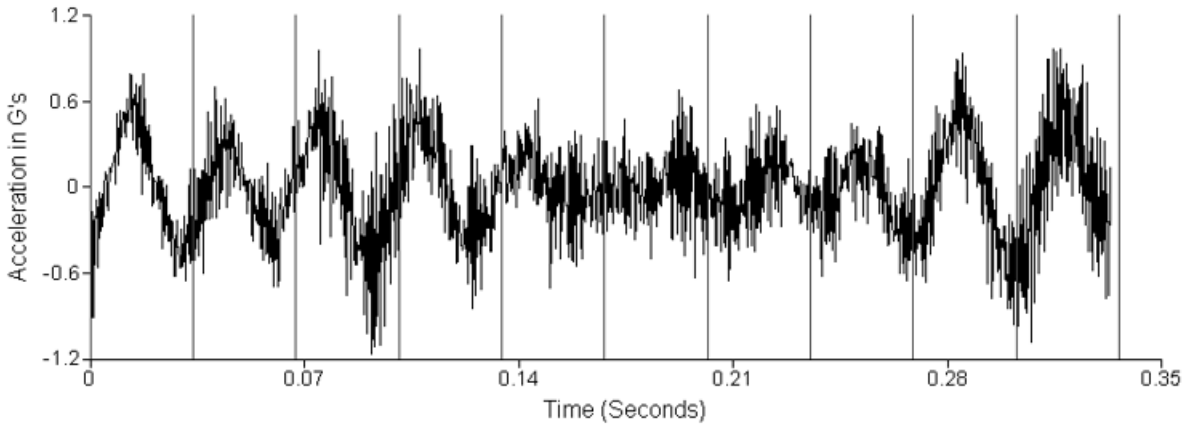
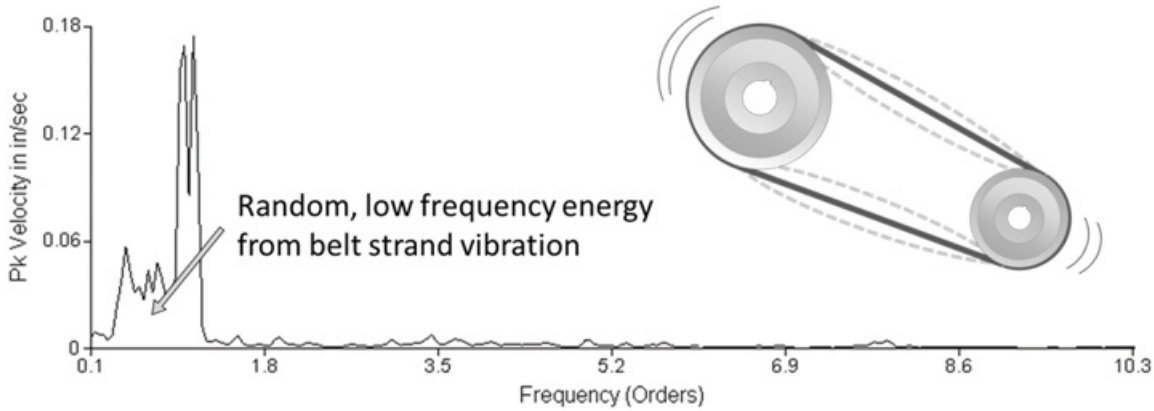
Waveform

Below is an example of a waveform measured on a belt driven fan that had a significant amount of belt strand vibration.



Spectrum

The spectrum below contains the random low frequency energy of the belt strand vibration.



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Summary

- Of the many belt designs, the V-belt is the most common used in industrial equipment.
- Belt operating frequency will always be sub-synchronous to the operating frequency of the sheaves in the system.
- A belt defect on a two sheave system causes vibration at 2 x belt frequency.
- Sheave misalignment results in 1 x RPM of the driver and/or driven component in the **axial** direction.
- An imbalanced or eccentric sheave causes vibration at 1 x RPM of the faulty sheave in the **radial** direction.

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Review Questions

1. Primary belt frequency is always what category of spectral energy?

2. What are some of the causes of belt vibration?

3. Name at least three variables that affect belt power transmission between the belt and the sheave.

4. A belt defect in a two sheave system will cause vibration at what frequency?

5. To help identify a belt defect in the spectral data, what can be done to the measurement set up?

6. Misaligned belts or sheaves tend to show higher vibration in which direction?

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Glossary of Terms

Appendix A

1/3 octave
Method of measuring a dynamic signal by measuring the signal levels within a set of bandpass filters that have a bandwidth of 1/3 octave.

Acceleration
An object's rate of velocity change. Usually measured in g's (or sometimes G's). G represents the acceleration due to gravity. 1 g = 386.4 in/s^2 = 32.2 ft/s^2. The international standard unit is m/s^2. 1 g = 9.806 m/s^2. The sensor used to measure acceleration is the accelerometer.

Accelerometer
The sensor used to measure acceleration.

Aliasing
An effect that can result in erroneous frequency spectra when a frequency within the signal being measured is more than 0.5 times the sampling rate.

Alignment
Positioning two or more components so that the rotational centerlines of their shafts are collinear at the coupling point under operating conditions.

Amplitude
The magnitude (RMS, peak, peak-to-peak, average, or dc) of a measured signal.

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Amplitude, Peak-to-Peak
The extent of a vibratory movement between its extremes; total excursion.

Amplitude, Peak
The extent of a vibratory movement from a mean (zero) position to an extreme.

Amplitude Modulation
Variation in the amplitude of a signal with respect to time.

Amount of Imbalance
The weight required in one or more planes to affect coincidence of an object's mass center line and rotational center line; usually given as the product of the weight and the radius at which it must be located;(i.e., oz.-inches).

Analog Integration
A method of converting from acceleration to the velocity signal or converting a velocity signal to the displacement signal. The time waveform is integrated and is converted to the spectra amplitude units.

Analysis Parameters
Divides the frequency spectrum into bands that are individually measured, analyzed, and trended.

Analysis Parameter Sets
Includes up to 12 individual analysis parameters, and also contains instructions that tell the machinery analyzer how to acquire data.

Angularity
The angle between two machine shaft centerlines. This angle is the same at any point along either centerline.

Autoranging
The process of automatically adjusting the input gain of an analyzer to match the amplitude of a signal. Optimizes the use of the dynamic range of the analyzer and improves signal-to-noise ratio.

Averaging
A method of collecting data where the spectra are averaged together. Typical methods are normal, peak hold, synchronous time, order tracking, and negative linear averaging.

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A-weighting
A frequency spectrum shaping that is applied to frequency spectra in acoustics. The effect is designed to approximate the way that the human ear perceives the loudness of sound. Sound levels are reduced at low frequencies and at very high frequencies where the ear is less sensitive. There are national and international standards for a-weighting.

Axial Float/End Float
Movement of one shaft along its center-line due to the freedom of movement permitted by journal bearings or sleeve bearings. This should be set before vertical and horizontal moves are performed.

Backlash
Refers to the torsional play caused by the design or degraded condition of a flexible coupling. This term is also used in gearboxes to describe the amount of torsional play between gears.

Balance
The condition which exists in a part such that no vibratory force is transferred to the supporting bearings of a part due to centrifugal force; the mass center line and rotational center line of a rotor are coincident throughout the rotor.

Band Pass Filter
A filter that attenuates frequencies above and below a specified range, thus allowing a certain range of frequencies to pass through unattenuated.

Band Rejection Filter
A filter that attenuates a specified range of frequencies and lets the frequencies above and below this range pass unattenuated.

Bandwidth
The analysis frequency range over which data will be collected. For normal route measurements, this is listed by the "BW =" parameter located on the screen display. It can be specified as a frequency range in Hz, as an order-based analysis (for example 10 x RPM), or in CPM. This range is often described as Fmax. This can also be used to describe the width of each line of resolution. BW = Fmax/# lines

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Baseline
A value stored as a reference, usually from the first data collected on a measurement point.

Baseplate
The surface, often made of steel plate or cast iron, to which the feet of a machine are attached. Usually mounted on foundation

Beat Frequency
The periodic change in vibration resulting from superimposing (adding) two different frequencies; the rate of increase and decrease in signal level dependent on the difference in frequencies (F1-F2).

Blade Passing Frequency
The number of blades multiplied by the rotational speed; additional frequencies may be related to the blade passing frequency by integer multiples, such as the number of stator blades near the rotating blades.

Bode Plot
A graphic plot that shows how the 1 x RPM amplitude and phase have varied with the RPM of a machine. These are always measured over a startup or coast-down of a machine and are used to identify shaft resonances and other signal characteristics.

BPFI
For rolling element bearings, the ball pass frequency (inner), caused by the rollers contacting a defect in the inner race of a bearing.

BPFO
For rolling element bearings, the ball pass frequency (outer), caused by the rollers contacting a defect in the outer race of a bearing.

Broadband Energy
Spectral energy in a broad range; usually caused by random energy in the time domain.

BSF
For rolling element bearings, the bearing ball spin frequency, caused by a defect on the roller contacting a single race of a bearing.

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Calibration
The procedure by which instruments and transducers are checked and adjusted in order to obtain accurate readings.

Cavitation
The formation of vapor bubbles in a liquid caused by a local pressure drop; the collapse of these vapor bubbles when the pressure increases can cause severe damage to the machine.

CMMS
Computerized Maintenance Management System

Coherence
A function of frequency which describes the degree of linear relationship between two signals. Used to assess cross-channel measurement quality, locate noise sources, and to understand or analyze transmission paths.

Collinear
Two or more lines positioned in space with no offset or angularity between them.

Coplaner
Lying or acting in the same plane.

Corrosion
The gradual destruction and/or pitting of a metal surface due to chemical attack. This chemical attack may be, but is not necessarily, due to the formation of acidic materials in the lubricant.

Coupling
Mechanical device that connects two or more shafts together. Could be a flexible coupling, fluid coupling, belt drive, etc.

CPM
Cycles per minute. 60 cycles per minute (CPM) is equivalent to one (1) cycle per second which equals one (1) Hz.

Crest factor
The ratio of peak to RMS levels of a signal.

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Critical Speed
The RPM at which a rotating part excites its own natural frequency.

Cursor
A manually controlled marker that can be moved across the plot display indicating frequency and amplitude at the cursor location.

Damping
A dissipative force that resists or reduces the movement of an object.

Damping Coefficient
Represents the amount of damping present in a system or subsystem.

Damping Ratio
The amount of damping present divided by the critical damping; indicates the significance of the effects of the damping on the system.

DC
Direct current.

Decibels (dB)
A logarithmic system of non-dimensional units that measures the size of a quantity relative to a reference level. Any quantity can be measured in this way, as can any two quantities with the same dimensions be compared using decibel measure. Given a reference pressure, Pref, any other pressure quantity P, having the same dimensions, may be expressed in decibels using the formula: If a quantity P is in RMS amplitude units, and Pref is a suitable reference level, the formula may then be rewritten using to give: $dB_{re\ Pref} = 20 \log_{10} (P/Pref)$

Demodulation
The process of extracting lower frequency data that is being modulated by higher frequencies via a process of filtering. See Envelope Detection.

Diagnostics
The techniques by which machine problems are identified and analyzed.

Digital Integration
A method of converting acceleration to velocity or velocity to displacement by first calculating the spectral data and then digitally converting the spectral peaks at each frequency. The time waveform is not integrated and is left in the sensor units.

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Displacement
Refers to the distance that an object has moved, usually measured in mils or microns. Displacement is often measured from eddy current probes and represents the physical movement of a rotating shaft. Sometimes accelerometers or velocity probes are used, and the data is integrated into displacement. In this case, movement represents the relative displacement of the machine casing where the probe is mounted.

Eccentricity (in Balancing)
The distance between the center of rotation and the center of mass.

Envelope Detection
Amplitude demodulation. A technique used to gain information about low frequency events which may be modifying high frequency components in a signal; the complex signal is band-pass filtered in some range of interest and then passes through circuitry to extract the instantaneous variations in the amplitude about its average value.

Epicyclic
Complex gearbox where the input and output shafts are inline.

Fast Fourier Transform (FFT)
A highly efficient mathematical procedure used to compute the frequency domain components from a time domain signal.

Fault Frequency Set
A set of pre-defined frequencies where mechanical faults are expected to occur.

Filter
A device for separating a complex wave into its different frequency components; a fixed band filter attenuates frequencies below and above the filter frequency limits; a variable frequency filter can be tuned through a frequency range, and it attenuates frequencies below and above the filter bandwidth to which it is tuned.

Final Balance Weight
The specific weight that will reduce the unbalance to within acceptable limits when applied to a known position.

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Flexible Rotor
The state of a rotor when its operating speed is more than 75% of its lowest natural frequency or bending mode.

Flutter
Vibration caused by the movement of fluid over a blade.

Flux Coil
The Flux Coil is a device which attaches to electric motors to capture flux signals. This provides an electrical quality signature which is sensitive to conditions that alter the electrical characteristics of the motor (broken rotor bars, eccentricity, imbalance between phases, and stator faults).

Flux Spectra
Motor flux spectra are acquired with Emerson's Flux Coil. The low frequency spectrum is a high resolution spectrum with a maximum frequency of at least three times running speed, or two times line frequency (whichever is greater). Analysis of this spectrum provides information on the rotor condition, changes in voltage balance and some stator fault detection. The high frequency spectrum contains slot pass family frequencies which provide indication of stator related faults.

Forced Vibration
Vibration of a system caused by an imposed force.

Foundation
The surface, often made of concrete, to which the machine baseplate is mounted, often with grout between baseplate and foundation to provide even support.

FPM
Feet per minute.

Free Vibration
Vibration without the presence of a driving force.

Fretting
A form of wear resulting from oscillating or vibratory motion of limited amplitude, resulting in the removal of very finely divide particles from the rubbing surfaces.

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Frequency
Number of times an event repeats in a specific period of time. Units are hertz (Hz equals cycles per second) CPM (cycles per minute), or Orders (cycles per rotation).

Frequency Demodulation
Determines the variation of the measured frequency from a carrier frequency of interest (typically, the average of the frequencies being measured).

Frequency Domain
Vibration data represented by a set of discrete composition frequencies, each of which has a characteristic amplitude; the output of a spectrum analyzer is a frequency domain representation of time domain signal.

Frequency Modulation
A process that encodes the amplitude variations of a complex signal by varying the instantaneous actual frequency from an average carrier frequency; the result is a high fidelity representation of low frequency signals in a high frequency format.

Frequency of Fibration
The cyclic movement occurring in a given unit of time.

Friction
The resistance to motion which is called into play when it is attempt to slide or roll one surface over another.

FTF
For rolling element bearings, the bearing fundamental train frequency; sometimes referred to as cage frequency. It is the frequency of the rotation of the cage and entire train of rolling elements.

Fundamental Frequency
A peak selected as the basis from which harmonic peaks are marked.

G's
A unit of acceleration, commonly used with the English system of units. One (1) g represents the acceleration due to gravity at sea level and is approximately equal to 386.4 in/s^2, or 32.2 ft/s^2 (9.806 m/s^2).

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Gear Dynamic Loads
Loads caused by the interaction of gear teeth while in motion; the variations in load that occur as each tooth engages and flexes generally result from manufacturing imperfections.

Gear Mesh Frequency
Synchronous vibration frequency associated with each pair of meshing gears. Calculated as: GMF(CPM) = shaft turning speed (RPM) • # of gear teeth

Grout
Mortar used for filling spaces between baseplate and foundation to provide even support.

Hanning Window
A shaping function applied to a time record before FFT is calculated in order to reduce leakage in the spectrum. Usually the default window type to use when analyzing continuous signals because of the compromise between frequency discrimination and leakage suppression.

Harmonic
An integer multiple of a fundamental frequency.

Harmonic Marker
A marker that appears on a spectral display to indicate the harmonic peaks of a fundamental peak.

Hertz (Hz)
A unit of frequency equal to cycles per second (CPS), usually abbreviated as Hz. One (1) hertz is equivalent to one (1) cycle per second, which equals 60 cycles per minute (CPM).

HFD
High frequency detection. The amplitude of vibration in G's over a broad frequency band from 5 kHz up to 20 kHz or greater.

High-Pass Filter
attenuates frequencies below a specified range.

Imbalance
Non-coincidence of the mass and rotational axes of rotor.

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Imbalance Moment
The resultant moment of all imbalance forces about the mass center of gravity; normally results in an angular displacement between the mass and rotational center lines.

Imbalance Vector
Plotted line on a polar chart that represents the force effect (amplitude and location) of imbalance.

Impacting
A very short term non-sinusoidal movement that often occurs as a result of excessive looseness in a machine and/or anti-friction bearing defects.

Impact Test
A type of test used to investigate the properties of a structure, in which the structure is caused to vibrate by an impulsive load from a hammer, and the vibratory response is measured by a vibration transducer.

Induced Soft Foot
A type of soft foot that is caused by external forces (coupling strain, pipe strain, etc.) acting on a machine independent of the foot to base-plate connection.

ISO
International Organization for Standardization.

Isolation
The reduction of the capacity of a system to respond to an external force by use of resilient isolating materials.

Jackscrew or Jackbolt
A bolt or screw attached to the base or foundation used to move or position a machine horizontally.

Jackshaft
A long shaft used as a spacer between two machines.

Lag-Angle
The angular position of unbalance relative to the position of maximum displacement of the bearing or shaft.

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Laminar Structure
A structure built up of layers of thin plates.

Lines
The number of lines of resolution used for the spectrum calculation. Resolution (in Hz/line) equals maximum frequency (in Hz) divided by the number of lines.

Lissajou
(ORBIT) Plot of the position of the center of the shaft in the bearing at various positions of angular rotation of the shaft, or as a function of time.

Low-Pass Filter
Attenuates frequencies above a specified range.

Micron(s)
A metric unit of measurement (one millionth of a meter).

Mil(s)
An English unit of measure for displacement (thousandths of an inch).

Mode Shape
The shape of the deflected shaft under a certain resonant condition.

Modulation
Instantaneous variations in a signal's characteristics. May be amplitude (AM) or frequency (FM) modulation.

Multiplane balancing
A method of balancing a rotor that allows the measurement and correction of the imbalance at more than l plane along the shaft of the machine.

Natural Frequency
The frequency at which a part or system resonates.

Nyquist Plot
A polar plot of the peak amplitude and phase of the 1 x RPM vibration component across a change in machine speeds.

Offset
Distance between rotational centerlines at any given normal plane, usually measured at the coupling midpoint.

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Oil Whip
Vibration of shaft typically caused by shaft deflections interacting with bearing fluid film effects such as oil whirl.

Oil Whirl
Source of vibration caused by effects of fluid film instability in journal bearings; typically, oil whirl produces vibration at frequencies of about 45% of the rotational frequency.

Operational Significance
This term is used to identify in importance of a machine in the operation of the entire plant.

Order Tracking
A measurement of a signal from a machine whose speed is changing with time, showing the level of one or more orders as a function of machine speed or time.

Orders
Multiples of machine turning speed (1xRPM, 2xRPM, 3xRPM, etc.). Frequency divided by turning speed.

Overlap
A function of averaging that reuses a percentage of the first average as the first part of the second and subsequent averages

PdM
Predictive Maintenance.

Peak
The largest signal level seen in a waveform over a period of time. For sinusoidal signals, the peak signal level is always 1.414 times the RMS value of the signal level. For non-sinusoidal signals, the peak level is often larger than the result that this formula would produce.

Peak-to-Peak
The difference between the maximum and the minimum levels (positive or negative) in a signal over a given period of time. For a sinusoidal (single frequency) signal, the peak-to-peak level is always two times the peak level and 2.828 times the RMS level. For non-sinusoidal (multiple frequency) signals this is no longer true and there is no simple relationship between peak-to-peak, peak, and RMS levels.

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PeakVue™
Analysis technique developed by Emerson which involves the measurement and analysis of stress waves to determine bearing and gear condition.

Period
The time required for one complete cycle of a periodic signal.

Phase (True or Synchronous)
The angular relationship between a tachometer pulse and the vibratory high spot of a synchronous vibration.

Phase (Relative or Cross-Channel)
The angular relationship between the vibratory high spot of two vibration sensors.

Phase-Angle Lag
The angular position of force causing a movement of a bearing housing or shaft at the instant of maximum excursion caused by that force; degree of lag depends on speed, supports, and mass.

Pipe Strain
Similar to soft foot condition where attaching piping to a machine distorts its frame.

Plane
Designates one or more of the rotating elements of a machine that is to be balanced. A plane can be a weight plane, a correction plane, or a measurement plane where sensors are attached. Each plane lies perpendicular to the line that defines the axis of rotation.

Polar Plot
A type of graph that displays data in polar coordinates.

Precession
The movement of the shaft in a circular orbit around the center of the bearing (rather than the small orbits around the equilibrium position typical of vibration); forward precession occurs when the shaft orbit is in the same direction as the shaft rotation; reverse precession occurs when the orbit is in the direction opposite shaft rotation; forward precession is typical of oil whirl; reverse precession is typical of dry friction whirl.

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Predictive Maintenance
Technology of periodically monitoring the actual condition of machines to discover faults, to determine probable time of breakdown, and to provide scheduled downtime for repair that avoids excess cost and lost production.

Pre-Trigger
Triggered data acquisition using a method such that the time record starts before the trigger event.

Radial Clearance (in Journal Bearings)
The clearance between the shaft and bearing when the shaft is centered in the bearing; equal to one half of the difference in diameters.

RBM™
Reliability-Based Maintenance®

Reference Mark
Any diagnostic point or mark which can be used to relate a position during rotation of a part to its location when stopped.

Reliability-Based Maintenance®
Incorporates elements of preventive, predictive, and proactive maintenance techniques. A total solution approach to maintaining plant equipment developed by Emerson.

Repeatability
The consistency (or variation) of readings and results between consecutive sets of measurements.

Residual Imbalance
The amount of imbalance which remains after the rotor is balanced.

Resolution
The frequency range represented by one line of an FFT spectrum. Calculated by dividing the maximum analysis frequency by the number of lines. The resolution in Hz is equal to the inverse of the data record length in seconds.

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Resonance
The response condition of a mechanical system where vibration input into the system is naturally amplified. The resonance frequency is determined by the mass, stiffness, and geometry of a system; the amplitude of the vibration that occurs at resonance is determined by the damping of the system.

Resonant Frequency
The frequency at which resonance occurs.

Rigid Rotor
Used to describe a rotor that normally operates at a speed 75% below its lowest natural frequency.

RMS
Root mean square. When applied to a dynamic signal such as vibration or sound, refers to an averaged level of a function obtained by averaging the square of the signal level over a period of time (or number of data records), then calculating the square root of the result.

Route
One or more machines and their respective measurement points organized in an efficient sequence for data collection.

RPM
Revolutions per minute.

Seizure
The stopping or prevention of movement of a mechanism as the result of interfacial friction.

Shaft Current/Voltage
Variations in electric motor air gaps, windings, and metal (rotor and frame) result in a fluctuating magnetic field which is cut by conducting material, in this case, the motor shaft. This condition gives rise to a current loop consisting of the shaft, bearings, and casing which can damage the motor shaft and bearings.

Shaft Probe
This device is used to acquire shaft current/voltage measurements on electric motors.

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Shim
A thin piece of material (usually metal), inserted between the base and machine feet to produce precise vertical adjustments of the machine center-line.

Sideband
A frequency component of equally spaced peaks around a center frequency that represents the effect of modulation on a signal. If a modulated signal has more than one component, each component will show sidebands. A sideband is spaced off from the frequency of the modulated signal by an amount equal to the modulating frequency. If the modulating signal has multiple components or if there is frequency modulation, the sideband pattern may be very complicated including sum and difference frequencies between the sideband component frequencies (intermodulation effects).

Single-Plane Balancing
Addition of balancing weight in a single plane to achieve coincidence of the mass center with the rotational center line.

Spectrum (or spectra)
The frequency domain representation of a signal. In practical measurements, the spectrum is usually displayed as a plot of magnitude verses frequency over a limited frequency range.

SST
Slow Speed Technology. This technique enhances low frequency spectral data and is very useful for machines with very slow turning speeds.

Steady-State Vibration
Condition induced by a continuing periodic force; usually referred to rotation or reciprocation of recurring nature.

Stiffness
The ratio of the deflection or movement of a part as related to the force applied.

Subharmonics
Vibration frequencies which are integer fractions of the running speed (example 1/2 RPM, 1/3 RPM, etc.) or some other fundamental frequency.

Subsynchronous Vibration
Vibration at any frequency lower than the rotational speed of the machine.

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Synchronous Vibration
Vibration at the rotational speed of the machine or some integer multiple of the speed.

Tachometer
A device that generates a pulse signal corresponding to the revolution of a shaft; used to measure turning speed. A single pulse per revolution may be used to trigger data acquisition synchronously with shaft rotation.

Transient
A non-steady-state signal of brief duration; often refers to a startup or coast-down of a machine.

Thermal Growth
Movement of shaft centerlines associated with (or due to) a change in machinery temperature between the static and operating condition.

Time Domain
Represents the amplitude of a signal as a function of time.

Torsional Play
The relative rotation between two coupled shafts that will cause the fixtures to move out of phase.

TIR
Total Indicator Runout. The total movement in mils that an indicator would read after the shaft is rotated 180° or 360°.

Torsional Critical Speed
Speed at which a twisting type of resonance occurs.

Transducer
a device that converts physical movement into an electrical signal proportional to the amplitude of the measured movement; (e.g., an accelerometer generates an electrical signal proportional to the acceleration of the point at which it is mounted).

Transient Vibration
A temporary or changing vibration or movement of a mechanical system, such as that caused by an impact or a speed change.

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Trend
Plotting a number of measurements of a parameter over time.

Trial Balance Weight
Added to a system to cause a change in vibration level or the position of the resulting unbalance force, or both.

Tribology
The science and technology of interacting surfaces in relative motion, and of related subjects and practices.

Trigger
Causes the machinery analyzer to start data collection upon the receipt of a specified dynamic signal from a sensor or a pulse from a tachometer.

Turbulent Flow
A type of fluid flow in which there is unsteady motion of the particles, the motion at a fixed point varying in no definite manner.

uniform Window
Analyzing a signal without shaping; no window is applied. Sometimes used for collection of non-steady-state data.

Velocity
The rate of change of displacement of a mechanical system. Units are inches per second (in/s or ips) in English units and m/s, cm/s, or mm/s in SI units. Can be measured directly with a velocity sensor or by integrating an acceleration signal from an accelerometer.

Vibration
Movement of a body about its reference.

Waveform
Analog or digital representation of a signal or function displayed as amplitude vs. time.

Wear
The removal of materials from surfaces in relative motion, normally by abrasion, adhesion or corrosion.

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Wedge Shim
Use of several shims to fill the wedge-shaped gap of a bent foot. Each shim is inserted to a different depth so that a stair-step shaped support is built to better support the entire foot. All of the steps together are called a step shim.

Window
See Hanning Window <XREF>and Uniform Window.<XREF>

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